Copyright, 1925, by the AMERICAN SOCIETY OF NAVAL ENGINEERS.

JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXXVII. AUGUST 1925 No. 3.

The Society as a body is not responsible for statements made by individual members

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published).

Rear Admiral C. F. HUGHES, U.S.N. — Rear Admiral B. C. BRYAN, U.S.N.
Captain W. T. CLUVERIUS, U.S.N. — Captain Q. B. NEWMAN, U.S.C.G.
Commander S. M. ROBINSON, U. S. N. — Lieut.-Comdr. E. L. COCHRANE, (C.C.) U.S.N.
Mr. J. F. METTERN — Mr. H. M. SOUTHGATE
Commander BRYSON BRUCE, U.S.N.

U. S. S. *DOBBIN* AND U. S. S. *WHITNEY*.

DESCRIPTION OF HULL AND MACHINERY INSTALLATIONS.

By LEWIS HOBART KENNEY, M. E., CIVIL MEMBER.

The *Dobbin* and *Whitney* were designed to meet all ordinary demands of service, supplies and repairs for a group of eighteen destroyers for a period of 60 days under wartime conditions. The necessary shop facilities for this duty are installed and include pattern, foundry, general machine, electrical, gyro, forge, coppersmith, sheet metal, ship carpenter, torpedo and optical shops. The supplies carried include fuel oil, lubricating oil, fresh water, provisions, spare parts of machinery, etc., etc. The medical facilities include a ward, an operating room, a laboratory, a contagious ward, a prophylactic room and a dental department

The construction of Destroyer Tender No. 3, the *Dobbin,* and Destroyer Tender No. 4, *the Whitney,* was authorized by act of Congress the 29th of August, 1916, and the building of the ships assigned to the Philadelphia and Boston Navy Yards respectively, on the 9th of July, 1919, and 9th of December, 1919.

	Dobbin	*Whitney*
Keel laid	23 December, 1919	23 April, 1921
Launched	5 May, 1921	12 October, 1923
Construction stopped due to lack of funds	July, 1921, to January, 1922	none
Commissioned	23 July, 1924	2 September, 1924
Official trials	28 to 31 October, 1924	not yet held

The following is quoted from "Names Selected for Naval Vessels" published in "Ships' Data, U. S. Naval Vessels:"

Destroyer Tender No. 3—*Dobbin*—named in memory of the late Hon. James Cochrane Dobbin, who was Secretary of the Navy from 1853 to 1857. He was born in Fayetteville, N. C., in 1814. In 1832 he was graduated from the University of North Carolina, and after being admitted to the bar in 1835 he practiced in Fayetteville. He was a member of Congress from 1845 to 1847, and also of the North Carolina Legislature from 1848 to 1852. In 1853 he became Secretary of the Navy, and during his administration he set about reform in all its branches. He was a firm believer in a strong Navy, and regarded the increase of naval strength not as a war but as a peace measure, and during his administration there were built 18 of the finest ships of their class that there were in the world. He instituted the present apprentice system, the inauguration of a retired list for officers unable to perform active duty, the law for increased pay to seamen, and honorable discharges for good conduct. When he retired from office he left behind him six first-class steam frigates, authorized upon his suggestion and constructed under his direction, for he gave them his personal attention in every detail. Under his auspices the Perry

Expedition was carried to a successful termination and the treaty with Japan made. He died at Fayetteville, August 4, 1857.

Destroyer Tender No. 4—*Whitney*—named in memory of the late Hon. William Collins Whitney, who was Secretary of the Navy from 1885 to 1889. He was born in Conway, Mass., July 5, 1841; was educated at Yale and Harvard and settled in New York City, where he was admitted to the bar. As corporation counsel of the city of New York in 1875-1882 he so completely reorganized and simplified the work of this office that thousands of dollars were saved annually. In 1885 he became Secretary of the Navy, and was a powerful advocate of naval expansion. He desired that the warships of the United States should be equal to the best in the world, and under his administration great progress was made in building the "new Navy." Two battleships, the *Maine* and the *Texas*—were authorized; also one armored cruiser, four gunboats, one practice vessel, one ram, one torpedo boat, and one dynamite gun cruiser. Mr. Whitney changed the Washington Navy Yard to the "Naval Gun Factory," confining its duties to the manufacture of ordnance. He died in New York, February 2, 1904.

The designs of the *Dobbin* were developed by the Philadelphia Navy Yard from type plans and specifications prepared by the Bureaus of the Navy Department interested in the construction of the ship, with the exception of the main turbines and reduction gears, which were developed from data and consulting services furnished by the Parsons Marine Steam Turbine Co., Ltd.

These plans were used in the construction of the *Whitney* where applicable. Where equipment of different characteristics was to be installed in the *Whitney* the Boston Navy Yard prepared the special plans for the work. For instance, the evaporating plants of the two ships are quite different and each Yard prepared its own installation plans. The pattern work and foundry work for the manufacture of the turbines and

reduction gears for the two ships was divided between the two Yards in accordance with a mutual agreement duly approved by the Navy Department, except the pinions and gear wheels of the main reduction gears which were manufactured by the Philadelphia Navy Yard for both ships. A similar arrangement applied to other material.

Considerable of the equipment such as pumps, machine tools, etc., was furnished by the Navy Department from excess stock of material which was purchased during the late war, for installation in ships then under construction but later scheduled to be scrapped in accordance with the conditions of the Washington Arms Conference.

In this article, unless special reference is made to the contrary, the information is applicable to the *Dobbin* and *Whitney*, and the author acknowledges the assistance of Mr. S. J. Egan and Mr. A. E. Russell of the Boston Navy Yard in checking the manuscript and providing the data applicable to the *Whitney*.

GENERAL DESCRIPTION OF HULL.

The construction of the ships was delayed in order that there might be incorporated into their designs the experience gained during the late war, in watertight subdivisions of the hull to resist torpedoes or mines; in the characteristics of supplies to be carried; and in the facilities which should be provided to meet the requirements of the service under wartime conditions.

HULL DATA.

Length, between perpendiculars, feet and inches	460–0
over all, feet and inches	483– 9½
Beam, moulded, feet and inches	60–10⅛
extreme, feet and inches	60–11⅝
Draft, forward, feet and inches	19– 9½
aft, feet and inches	22– 3½
mean, feet and inches	21– 0½
Displacement L. W. L. (21 feet-0½ inch-draft) tons	10,600
Tons per inch immersion at L. W. L.	49.8

Area of amidship section, square feet	1224
Area of L. W. L., square feet	20,870
Area of wetted surface, square feet	35,040
Coefficient of fineness, block	.630
midship section	.952
L. W. L.	.736
Ratio of length to beam	7.54

HEIGHTS ABOVE THE 20-FOOT-0½-INCH L. W. L.

Top foremast, feet and inches	171– 1
mainmast, feet and inches	171– 2
Bottom of floor of crow's nest, feet and inches	86–11
Pilot house, feet and inches	42– 3
Searchlight platform, forward, feet	63
aft, feet and inches	41–11
Floor of standard compass stand, feet and inches	49–11

TANK CAPACITIES.

Fuel oil, service, tons	958*
cargo, tons	1415*
Lubricating oil, tons	91*
Gasoline, tons	70*
Alcohol, tons	17*
Reserve feed water, tons	305
Fresh water, tons	199

TANKS, NUMBER OF.

Fuel oil, service	24
cargo	47
Lubricating oil	4
Gasoline	2
Alcohol	1
Reserve feed water	5
Fresh water	8

A brief description of each deck is as follows:

TOP OF BRIDGE DECK HOUSE.

This deck extends from Frame 48 to Frame 58 and on it are installed a 12-foot range finder, a range repeater and two torpedo directors.

* 95 per cent capacity.

THE BRIDGE.

This deck extends from Frame 46 to Frame 82. On the bridge are located the steering stand, magnetic and gyro compasses, engine and steering telegraphs, voice tube terminals, loud speaking telephone and interior communication system apparatus, two pelorus stands, above which are mounted one meter range finders, side lights, an emergency cabin and chart house, and aft of the bridge are located life rafts, vegetable lockers, radio compass house and anti-aircraft guns.

THE BRIDGE DECK.

This deck extends from Frame 47 to Frame 83. On this deck is the chart and pilot house, captain's quarters, the force commander's office, staff officers' staterooms and baths, the optical shop, radio rooms, sounding machines, 6-pounder saluting guns, Franklin life buoys and whaleboats.

SUPERSTRUCTURE DECK.

This deck extends from Frame 47 to Frame 153. On this there are installed the force commander's quarters, staterooms and baths of the officers, and the ward room and staff's messroom, four 5-inch guns, boats, the winches for handling boats, vegetable lockers, anti-aircraft guns and searchlights.

MAIN DECK.

This deck extends the full length of the ship. On this are installed the anchor handling gear, four 5-inch guns, winches for cargo booms, warrant officers' staterooms, mess room and washroom, dental office, officers' staterooms, ward room and washroom, executive officer's office, officers' galley, foundry, coppersmith shop, blacksmith shop, shipfitters shop, ship carpenters shop, incinerator room, battery charging room, searchlight rheostat room, bakery, powder testing room, butcher shop, crew's galley, sick bay, towing engine. Outside of the deck houses are located the cylinders containing CO_2, oxygen and acetylene.

SECOND DECK.

This deck extends the full length of the ship and on it are forward storerooms, post office, supply officer's office, the shop office from which the work of the several shops is directed, the general machine and electrical shop, pattern shop, general mess pantry, firemen's washroom, C. P. O.'s quarters and crew's quarters, gyro repair shop, laundry, central telephone station, engineer officer's office, general mess issuing room and steering gear room.

THIRD DECK.

This deck extends the length of the ship and forward are storerooms, crew's space, torpedo work shop, cold storage rooms for cargo and the ship's service purposes, ice machine room, various storerooms, including engineer's storerooms; aft are storerooms, also a barber shop, printing shop, tailor shop, shoe repair shop and prisons.

PLATFORM DECK.

This deck extends from the bow to Frame 57 and from Frame 126 aft. Forward is the chain locker and aft of it are the various magazines for the stowage of ammunition, torpedoes and so on. Outboard are tanks for the stowage of service and cargo fuel oil and aft are storerooms.

INNER BOTTOM.

In the inner bottom is stored service fuel oil, and reserve feed water.

COMPLEMENT.

The ship's complement consists of the following:

Force commander	1
Staff officers	7
Commanding officer	1
Ward room officers	19
Warrant officers	10

	Ship	*Repair*
Chief petty officers	20	16
Seamen's branch	122	16
Machinery branch	12	28
Engine room force	54	67
Auxiliary branch	15	—
Special branch	35	10
Messmen's branch	21	—
Total	279	137
Total	416	

SMALL BOATS.

2—40-foot Motor launches	capacity each	90
4—36-foot Motor launches	capacity each	70
6—20-foot Motor launches	capacity each	19
2—35-foot Motor boats	capacity each	27
2—30-foot Whaleboats	capacity each	40
2—16-foot Dinghies	capacity each	10
2—14-foot Punts.		
2—Lifeboats 9 feet × 14 feet	capacity each	67
2—Lifeboats 7 feet × 12 feet	capacity each	41
4—Lifeboats 5 feet × 10 feet	capacity each	20
Total		1024

BATTERY.

The main battery consists of 5-inch breech loading rifles. Four are located on the main deck and four on the superstructure deck. Four 3-inch anti-aircraft guns, one located forward amidships on the main deck, two aft of the bridge, and one aft on the superstructure deck. There are two 6-pounder saluting guns on the bridge deck.

Two torpedo tubes are located in the torpedo work shop primarily for testing out torpedoes which have been undergoing overhaul.

BATTERY DATA.

8—5-inch Mount, Mark XV, 51 Cal. B. L. R.'s.
4—3-inch Anti-aircraft Mounts, Mark XI, Mod. 2, 50 Cal. B. L. R.'s.
2—6-pounder Mount, Mark X, Saluting guns.
2—21-inch Single Torpedo Tubes, Mark VII.
1—12-foot Range Finder, Mark XVII.
2—1-Meter Range Finders, Mark XIV.
2—Torpedo Directors, Mark XI, Mod. 4.
1—Range Keeper, Mark II.

LAUNDRY.

The laundry is located on the port side between Frames 104 and 122 on the second deck. The following equipment is installed:

42-inch × 72-inch Washer:—Henrici Laundry Mach'y Co.
2—30-inch Extractors:—American Laundry Mach'y Co.
2—38-inch Universal presses:—1-American Laundry Mach'y Co., and 1-Prosperity Co.
1—Flat work ironer:—*Dobbin*, American, and *Whitney*, Troy Laundry Mach'y Co.
1—Collar shaper:—American Laundry Mach'y Co.
1—Disinfector:—American Sterilizer Co.
1—Ironing board:—*Dobbin*, Phila. Navy Yard, and *Whitney*, American Laundry Mach'y Co.
1—Wash tub:—American Laundry Mach'y Co.
1—Starch cooker:—American Laundry Mach'y Co.
1—3-Drawer drying room:—American Laundry Mach'y Co.
1—Tumbler dryer:—American Laundry Mach'y Co.
2—Truck tubs.
1—Portable table.
4—6½-pound Electric irons.
2—8-pound Electric irons.

The forward portion of the laundry space is tiled on account of the water from the washer, extractor and other equipment installed therein. A coaming separates this portion from the after portion where the dry work is done.

Aft is a receiving room which is fitted up with shelves and pigeonholes to facilitate sorting of the laundry, and the end of the disinfector projects into this room into which the clothing to be disinfected is introduced, the disinfected clothing being discharged into the main laundry room. Ample facilities are provided for taking care of the ship's force as well as the laundry of those on board ships which are undergoing repair by the ship.

GENERAL MESS PANTRY.

The general mess pantry is located between Frames 90 and 94 on the starboard side, second deck, and in the room is installed a dish washing machine, sinks, dressers, and racks for the dish washing machine baskets. The floor of this room is tiled.

CREW'S GALLEY.

The crew's galley is located between Frames 124 and 128 on the main deck, and extends across the full width of the deck house. In the galley are installed three 80-gallon, one 60-gallon and one 40-gallon steam jacketed kettles, a three-section range, dressers and sinks, a cake machine, wash basin, etc. The ranges are designed for oil burning, fuel oil being supplied from the ship's system. Atomization is obtained by means of compressed air for which a special air compressor is installed. Steam for atomization is also available. The floor of the galley is tiled.

OFFICERS' GALLEY.

The officers' galley is located between Frames 86 and 90 on the main deck on the starboard side of the deck house. In the galley are installed a five-section oil burning range, dressers and sinks and a wash basin. The ranges are equipped for oil burning. The floor in this compartment is tiled.

BAKERY.

The bakery is located between Frames 119 and 124 on the main deck, starboard side of the deck house. In the bakery are located yeast tub, wash basin, dough mixer, three oil burning ovens, sink, steam proofer, and dough troughs. In a corner of the room, but separated from it, is a potato peeler. The floors of these rooms are tiled.

ANCHOR HANDLING GEAR.

The windlass was built by the Mare Island Navy Yard and is of the steam driven type, equipped with the usual brakes, clutches, and so on. At the forward end are two gypsy heads which can be used if the wildcats are secured. The ship carries three bower anchors of the stockless type, one being a spare. The usual devil claws and so on are provided.

Cylinders, number	2
diameter inches	12
Stroke, inches	16
Crankpin, diameter, inches	3
length, inches	3½
Connecting rod, length between centers, feet and inches	5– 4⅝
Crosshead pin, diameter, inches	2¼
length, inches	2¾
Steam nozzle, diameter, inches	4
Exhaust nozzle, diameter, inches	4½
Designed operating steam pressure, pounds	125
Gear train windlass	10 by 48 by 10 by 60 teeth
Gypsy heads	10 by 48 by 28 by 28 teeth
Hoisting speed with two anchors, fathoms per minute	6
Size of anchor chain, Navy Standard stud link, inches	2¾
Kedge anchor, number	1
weight, pounds	1500
Stream anchor, number	1
weight, pounds	3000
Bower anchors, number	3
weight, pounds	14,000

STEERING ENGINE.

The steering engine was designed and built by the Philadelphia Yard for the *Dobbin* and *Whitney*. It was designed for steam operation, two cylinders being used, the cranks being set at 90 degrees. The engine drives through gearing a right and left hand screw operating connecting rods which actuate the rudder. The steering engine by a suitable clutch may be disengaged if required and by another suitable clutch the hand steering gear can be engaged. Three steering wheels are provided for hand steering.

An adequate system of lubrication is installed throughout and a forced lubricating system is installed on the steering engine which lubricates the crankpin and the wrist pin bearings.

The rudder stock is so designed that the load or weight of the rudder is borne by a casting secured to the deck. The liner on this deck was bored and faced at the time the pintle bearings of the rudder were bored in order to assure accuracy of alignment. On this liner is secured the casting mentioned above, which carries the weight of the rudder. A heavy brass ring is installed between the top of this casting and the casting secured to the top of the rudder stock, and a trough placed around this ring provides adequate lubrication at all times.

Provision is made for controlling the steering gear by means of a trick wheel at the steering engine. A system of wire rope control and shafting and gearing connect to the steering stands in the chart house and bridge. The wire rope drum at the engine is of the Hanscom variety, the springs being of adequate strength to maintain proper tension in the cable system. Special guides to keep the rope in its proper drum-groove were installed. The dimensions of the steering engine are as follows:

Cylinders, number	2
diameter, inches	12
stroke, inches	12
Nozzle, steam, inches	3½
exhaust, inches	4

Crankpin, diameter, inches	3¾
length, inches	4½
Crosshead pin, diameter, inches	3
length, inches	3
Connecting rod, length between centers, inches	24
Transmission gears	16 by 47 teeth
Operating steam pressure, pounds	120
Time required to put rudder from hard over to hard over, seconds	30
Rudder, angle of train, degrees	70
area in square feet	200
stock, diameter, inches	15½
Napier screw, double Acme threads, lead, inches	1½
outside diameter, inches	7⅛
Connecting rods, length between centers, feet and inches	8– 5
Distance from center of rudder stock to connecting rod center, feet and inches	2– 4
Bearing surface to carry load of rudder, square inches	615

CARGO AND BOAT HANDLING GEAR.

On the foremast are located four cargo handling booms. The two booms forward of the mast are each 44 feet 6 inches long, and the two aft of the mast are each 44 feet 4 inches long. Four steam driven winches are located on the main deck near the foremast; they are single geared, single drum, reversible, with drum bolted to driving gear and are fitted with one gypsy head. The drum is equipped with a foot brake.

On the superstructure deck between Frames 86 and 88 is located one steam driven winch, single geared, single drum, reversible, with two gypsy heads. The drum is equipped with a foot brake.

On the after side of the mainmast are two 44-foot 6-inch booms, and on the superstructure deck are located two steam driven winches, single geared, single drum, reversible, with cone friction drive and two gypsy heads. Each drum is equipped with a foot brake.

DATA APPLICABLE TO EACH WINCH.

Manufacturer	American Clay Mach'y Co.
Steam cylinders, number	2
diameter, inches	9
stroke, inches	9

Nozzle, steam, diameter, inches	3
exhaust, diameter, inches	3½
Capacity	5000 pounds at 200 feet per minute over single whip, or 22,400 pounds at 35 feet per minute over 6 part purchase
Capacity of 44-foot 4-inch booms, pounds	24,000
44-foot 6-inch booms, pounds	16,000

AMMUNITION HOIST.

There are two electrically operated ammunition hoist winches manufactured by Lidgerwood Manufacturing Company, located on the main deck, one between Frames 32 and 34 and the other between Frames 45 and 47, each with a capacity of 1000 pounds at 100 feet per minute. The winches are single geared and each is fitted with a reversible drum and controller. The watertight motors are wound for 120 volts D. C. and are fitted with electro-magnetic brakes and the resistances are located below deck.

TORPEDO HOIST.

There is one electrically operated torpedo hoist winch manufactured by Lenher Engineering Company, located on the second deck between Frames 69 and 71, port side, with a capacity of 4000 pounds at 600 feet per minute. The winch is single geared and is fitted with a reversible drum controller. The drum controller is designed for dynamic braking. The semi-enclosed motor is wound for 120 volts D. C. and is fitted with an electro-magnetic brake.

TOWING ENGINE.

The steam driven towing engine, manufactured by American Engineering Company, is located on the main deck aft. Its purpose is to maintain a uniform tension on the towline at all times. A device is installed on the engine aft of the drum to guide the wire rope as it is coiled on or uncoiled from the drum. On the rail is located a towing chock to assure that the tow line runs free at all times.

By means of a system of clutches the engine can be used on towing or for operating two gypsy heads. The gypsy heads are located on the forward end of the engine, and the shaft on which the gypsy heads are located is of sufficient length to locate the gypsy heads where desired for hoisting purposes.

Steam cylinders, number	2
diameter, inches	16
stroke, inches	16
Nozzles, steam, diameter, inches	3½
exhaust, diameter, inches	4
Drum, diameter, inches	34
length, inches	40
capacity in feet of rope	1800
Rope, diameter, inches	2
Operating steam pressure, pounds	100
Maximum pull at 100 pounds steam pressure	33,800
Gear train for towing	12 by 66 teeth
Gypsy heads	21 by 45 teeth

INCINERATOR.

The garbage and refuse incinerator of the Jockers type, manufactured by the Navy Yard, New York, is installed on the main deck, starboard side, between Frames 100½–104. A fuel oil tank of 50 gallons capacity is installed in the room. Fuel oil for refilling the tank is supplied from the ship's system. The oil burning outfit is designed for air atomization supplied from the galley air compressor. However, steam atomization is provided for use in case of an emergency. A cast iron flue extends from the incinerator to the smokepipe to carry off the products of combustion. The charging platform is located on the superstructure deck, the charging of the furnace being done through an opening in that deck.

FRESH HOT WATER SYSTEM.

A central hot water heating system provides hot water throughout the ship. The heater is installed on the second deck near Frame 102, just aft of the smokepipe in the compartment around the smokepipe. The heating agent is auxili-

ary steam which is admitted to the heater through a thermostat control valve. The thermostat can be set to any desired temperature so that the water discharged from the heater is at this constant temperature. The condensate drains through a trap to the main feed and filter tank. The hot water is circulated through a loop system continuously and short extensions are led from the system to faucets. The extensions were made of minimum length in order to reduce as much as possible the amount of water drawn off before hot water is discharged from the faucet. Two motor driven centrifugal pumps, each of ample capacity to operate the system, are installed near the heater. Water is supplied to this system from the ship's gravity tank and automatically supplies water to replace that drawn off through the faucets. Faucets are located as follows:

Captain's pantry; Captain's bath; Staff Officers' bath; Force Commander's pantry; Force Commander's bath; Ward Room Officers' pantry; Dental office; Warrant Officers' pantry; Warrant Officers' bath; Ward Room Officers' bath; several rooms in the sick bay; C. P. O.'s washroom; Firemen's washroom; General mess pantry; Retiring room.

HEATERS.

Manufacturer	Row & Davis
Heater, number	1
Nozzle, exhaust steam inlet, diameter, inches	3
drain outlet, diameter, inches	1¼
water inlet, diameter, inches	2½
water outlet, diameter, inches	2½
Heating surface in square feet	20
Capacity in gallons of water per hour from 50 degrees F. to 180 degrees F.	1200

PUMPS.

Manufacturer	Morris Machine Works
Pumps, number	2
Type	motor driven, centrifugal
Nozzle, suction, diameter, inches	2½
discharge, diameter, inches	2½
H. P. of motor	2
Supply D. C. volts	120

R. P. M.	1700–1900
Gallons discharged per minute	40
Head, suction, feet	10
discharge, feet	20

FRESH WATER SYSTEM.

The ship's supply of fresh water is stored in the following tanks located between Frames 113–126:

Tank No.	*Gallons Capacity.*
C-14-W	9442
C-15-W	11394
C-16-W	4705
C-17-W	4705
D-2-W	4662
D-3-W	4662
D-4-W	7136
D-5-W	7136
Total	53,842
Gravity tank, gallons	1000

Water is drawn from the storage tanks by pumps located in the evaporator room through a suitable manifold system and discharged to the ship's system which terminates in a gravity tank located on top of the bridge deck house at Frames 60–62. An automatic cutoff valve located in the gravity tank shuts off the admission of water when the level has arrived at a predetermined height. Fresh water filling connections are provided on the port and starboard sides accessible from the main deck at Frames 37 and 122. Water can also be discharged from these filling connections for filling the tanks of other ships, or any similar service.

Water is piped to all pantries, all baths, all washrooms and all staterooms, also to the optical, foundry, blacksmith and general machine shop and shop office, powder test room, butcher shop, bakery, officers' galley, crew's galley, potato

peeler room, rooms in sick bay, laundry, scuttle butts, ice machine room, bake shop, printing office, and the engine and fire rooms. There are two scuttle butts, both located on the second deck. The forward one is in the general machine shop, Frames 52–54, and the aft one is in crew's quarters space, Frames 128–130.

FIREMAIN SYSTEM.

Water is supplied the firemain system from the fire and bilge pumps, one located in each fireroom and one in the engine room. The two distiller circulating pumps in the evaporator room can also supply water to the firemain. The firemain extends throughout the ship and fire plugs are located as follows:

Bridge deck, P. and S. Frame 65

Superstructure deck, P. and S. Frame 65
S. Frame 82
C. Frame 109
C. Frame 131

Main deck, P. and S. Frame 22
C. Frame 37
P. and S. Frame 65
S. Frame 82
P. and S. Frame 109
S. Frame 131
C. Frame 148

Second deck, S. Frame 19
P. Frame 37
S. Frame 57
S. Frame 82
P. Frame 104
S. Frame 130
S. Frame 144

Third deck, S. Frame 18
P. Frame 56
S. Frame 57
C. Frame 90
C. Frame 109
S. Frame 139

Platform deck, C. Frame 26
P. Frame 33
P. Frame 40
P. Frame 45
P. Frame 52

The firemain provides connections for flooding the magazines and to the flushing system.

FLUSHING SYSTEM.

Water is supplied the flushing system from pumps, one located in each boiler room and two in the engine room. A cross connection is provided to supply water from the firemain system with the usual safety valves and other protection features. The system extends to all baths, toilets and washrooms, and also to the crew's galley, potato peeler room, laundry, ice machine room and gasoline stowage tanks.

DRAINAGE SYSTEM.

The drainage system extends throughout the length of the ship and is arranged to pump out the peak tanks and the various compartments located above the inner bottom. Pumps for this service are located in each fireroom and in the engine room. Suction wells with perforated covers are let into the inner bottom plating for the suction inlets of the system. Valves located for convenient operation of the system are installed and certain of them are provided with deck operating control.

FIRE PROTECTION FEATURES.

In view of the amount of fuel oil and gasoline carried on board and the attendant fire risks, the question of fire protection was given very thoughtful consideration. Special escape trunks were installed in each fire room with openings at the floor plate level and exits on the third deck. There is a foam fire extinguisher installed in each fire room of the regular Navy standard type. In the engine room there is installed a foam fire extinguisher of the same type as installed in the fire rooms, and an escape is located above the evaporator room accessible from the dynamo platform level which opens on the superstructure deck. Located throughout the ship are carbon tetrachloride fire extinguishers of one quart capacity placed as follows:

Bake shop	2
Battery charging station	2
Boiler rooms	4
Blacksmith shop	2
Ship carpenter shop	2
Crew's galley	2
Officers' galley	2
Dynamo room	2
Engine room	2
Foundry	2
Lamp room	2
Laundry	2
Machine shop	2
Paint and oil room	2
Radio room	2
Shipfitters shop	2
For the use of the Fire and Rescue Party	48
Fire and Rescue Party (2½-gallon acid-soda type)	2

SERVICE FUEL OIL SYSTEM.

The service oil, or the oil provided for the use of the ship for its operation, is stored in the service oil tanks. Earlier in this article is given the number of service oil tanks and the total capacity for service fuel oil. The system is controlled from suction and discharge manifolds, one located in the forward end of the forward fire room and the other in the after end of the after fire room. These manifolds are cross connected. High and low suction valves are installed for each tank.

There are installed in the tanks heating coils to warm the oil if necessary to improve its viscosity for pumping. Steam to these heating coils is supplied from two control manifolds located above the two service oil manifolds. The tanks piped for this system are those in the inner bottom although certain of the outboard tanks are designated service fuel oil tanks. The oil from those tanks can be drawn through the cargo fuel oil system and discharged into the inner bottom service oil tanks. Extensions from the steam to heating coils lead to the tops of the tanks for steaming out.

CARGO FUEL OIL SYSTEM.

The number of tanks for carrying cargo fuel oil and their capacities are given earlier in this article. There is a pipe trunk on each side of the ship through which the main pipe line to these tanks is installed. Extensions from this line lead to the various tanks. Cutout valves are provided which are fitted for deck operation. The piping system terminates at the cargo fuel oil pump, of which one is located in the forward fire room and one in the after fire room. These pumps can discharge oil from any tank to any other tank, cargo or ship's service, within certain limitations, and overboard, as may be required. The port and starboard systems are cross connected. The fuel oil filling connections are provided at the ship's side accessible from the main deck as follows:

P. and S. Frame 40
P. and S. Frame 65
P. and S. Frame 87
P. and S. Frame 108

The filling connections can also be used as discharge connection when occasion requires. Extensions from the steam to heating coils lead to the tops of the tanks for steaming out. An overflow system is provided to protect the tanks from excessive pressure.

PNEUMERCATOR SYSTEM FOR FUEL OIL TANKS.

Gauges made by the Pneumercator Company are installed in the fire rooms for indicating oil levels in all service and cargo fuel oil tanks. The gauges for the service oil tanks are located above the service oil control manifolds. The other gauges are located near the manifolds of the cargo fuel oil pumps. A motor driven automatically controlled air compressor to provide compressed air for the operation of the gauges is installed in the after fire room.

In this system there is installed for each gauge a balance chamber near the center and at the bottom of each tank. The balance chamber is connected to the gauge by tubing. In order to make an observation of oil level the compressed air is admitted to the system until it bubbles from the balance chamber. The air pressure is then transferred to the mercury column, and the mercury column balances the weight of the liquid column in the tank. The scale of the gauge is graduated in accordance with the calibrated capacity of the tank and direct readings are obtained in accordance with this scale, which is graduated to read feet in depth and pounds of oil. The scale indicating in feet and inches is graduated for an average specific gravity of oil to be stored in the tank, and the purpose of this scale is to give a visual indication of approximately the depth of oil in the tank in terms of feet and inches, whereas

the scale graduated in pounds of oil in the tank indicates correctly and is independent of changes in specific gravity of the oil.

GASOLINE STOWAGE SYSTEM.

The capacity of the tanks for the stowage of gasoline is stated earlier in this article. The drawings for the system were developed by the Philadelphia Yard from data and specifications furnished by The Farr Hydraulic Systems, Inc., and duly approved by the Bureau of Construction and Repair. In this system the tanks are first completely filled with water and afterwards the gasoline is pumped in, displacing the water which flows overboard. In order to discharge the gasoline from the tank water is admitted under sufficient pressure for this purpose. The permissible water pressure, however, is such that no water will be discharged overboard from the gasoline pipe. The water for the operation of this system is supplied from the flushing system.

By keeping the tank completely filled there is no opportunity for the mechanical mixing of the gasoline and water. Furthermore, the development of free gases is prevented. The weight in the tank is practically constant, thus eliminating "free surface." Furthermore, this system assures the delivery of clean gasoline from the top of the tank.

The principle of operation of this system is indicated in Fig. 1. The control stands, one port and one starboard, are located on the main deck, between Frames 82 and 83. Each stand operates a valve located between the second and third decks. The movement of the operating lever either admits water to the tank or connects the tank to the overflow. The overflow so arranged provides a safety feature for the expansion of the liquids in the tank and the overflow of the water when the tank is being filled with gasoline. The port and starboard tanks were cross connected as indicated by the diagram so that gasoline could be discharged from either the port or the starboard tank, or either tank could be filled from the port or starboard sides.

In order to determine the amount of gasoline in a tank a cleavage gauge manufactured by the Pneumercator Company was installed. The gauge operates on the pneumercator principle but there are two balance chambers, one located near the

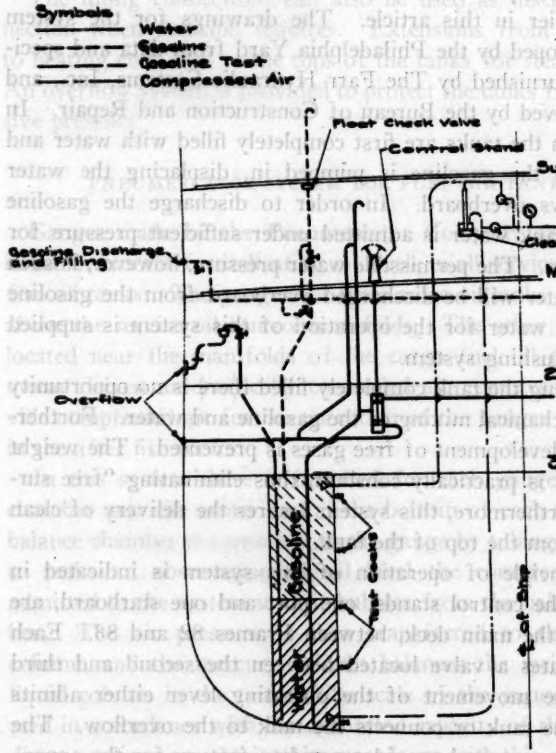


FIG. 1.

top, and one near the bottom of the tank as indicated in Fig. 1. To operate the gauge air is admitted to the lower balance chamber until it bubbles into the tank, and the same operation applies to the upper balance chamber. The column of mercury

then balances the weight of the liquid in the tank and the scale of the gauge is graduated for this condition. It is, however, necessary to know the specific gravity of the gasoline and the specific gravity of the water in the tank. These three readings applied to a slide rule provided with the outfit, make the determination of the amount of gasoline in a tank a relatively simple matter.

ALCOHOL STOWAGE SYSTEM.

The capacity of the alcohol stowage tank is stated earlier in this article. The tank is fitted with the usual filling and venting system, but in addition a T-fitting is installed on the top of the vent pipe. Into each end of the T-fitting is screwed a nipple and onto these nipples check valves so arranged that one valve permits the egress of air when the tank is being filled and the other the ingress of air when the tank is being emptied. This arrangement seals the tank and prevents evaporation of the alcohol, at the same time not interfering with the operation of the system. An alcohol pump is located in the engineers storeroom just above the alcohol tank and is piped to discharge through a valve in the storeroom and through a valve located on the main deck. The rate of discharge of the pump is adjustable from one pint to one gallon. The pump, furnished by S. F. Bowser & Co., Inc., is fitted with a lock to prevent any unauthorized use of the system.

MAGAZINE FLOODING AND SPRINKLING SYSTEM.

Provision is made for flooding the magazines through sea valves which may be operated from the main and second decks. The operating stations on the second deck are located in a passageway and the rods terminate in an illuminated box fitted with a glass front.

Provision is made for sprinkling the magazines from the firemain system. The group control valve may be operated

from the main, second and third decks. Where the pipes pierce watertight bulkheads in the magazine spaces check valves are installed to prevent flooding back.

The usual nameplates are installed to explain the purpose and location of the valves.

SCUTTLING SYSTEM.

There are located on the port and starboard sides below the load water line 20 scuttling valves, 10 on each side. The inlet diameter of these valves is 10 inches and the valves are arranged for operation at the second and third decks. Under normal conditions the discharge nozzle of the valves are covered with a blank flange.

CARGO LUBRICATING OIL SYSTEM.

The number and capacity of the cargo lubricating oil tanks are given earlier in this article. The usual filling and venting system is installed on all the tanks. In the engine room aft on the port side is located the cargo lubricating oil pump and its control manifold. An oil meter is installed to measure all lubricating oil discharged by the pump. The control manifold is of the double compartment type to permit the pumping of oil from one cargo tank and discharge it to any other cargo tank, or to the ship's side, or to the engine room lubricating oil tank located in the upper part of the engine room aft of Frame 110. The discharge overboard connections are accessible from the main deck and are located between Frames 108 and 109, port and starboard sides.

This line can also be used for filling the cargo lubricating oil tanks. The telltale or overflow pipe is installed in the after port side of the engine room to indicate when the tanks are filled, the overflow being discharged into a small tank provided for this purpose.

CARGO LUBRICATING OIL PUMP.

Manufacturer	Worthington Pump & Machinery Corporation
Cylinder, steam, diameter, inches	3½
oil, diameter, inches	4
stroke, inches	4
Nozzles, steam, diameter, inch, tap	⅜
exhaust, diameter, inch, tap	½
suction, diameter, inches	1½
discharge, diameter, inches	1½

LUBRICATING OIL TANKS.

Tanks for the storage of lubricating oil are located throughout the ship for the following purposes:

Windlass and forward deck machinery	10 gallon tank
Steering engine and towing engine	10 gallon tank
Machine shop tool room	50 gallon tank
Electrical shop	5 gallon tank
Ice machine room, lubricating oil	50 gallon tank
ice machine oil	250 gallon tank
Boiler rooms, each	5 gallon tank
Evaporator room, *Dobbin* only	5 gallon tank
Main engine room,	
Turbo generator oil, 2 tanks each, *Dobbin* only	250 gallon tank
Cylinder oil	50 gallon tank
Main engine lubricating oil, 4 compartment tank, total	2000 gallon tank

COMPRESSED AIR SYSTEM.

There are two steam driven air compressors and their reservoirs located in the after fireroom. There is one extension from these reservoirs leading to lines on the port and starboard sides, cross connected for the gas ejecting system. There is an extension to each 5-inch gun from this system, air being supplied through a Lalor automatic cutoff valve to a reservoir located directly below the gun. The function of the Lalor valve is to automatically close if the pipe line on the discharge side of the valve should become broken.

Another pipe line runs from the reservoir tanks for the ship's system with leads to the several shops, to the galley system, and to the air testing stations. There are eleven air test

stations from which air can be obtained for testing compartments. The piping system terminates in a relief valve and a hose valve, also a pressure gauge. Portable apparatus contains an additional reducing valve and a relief valve and mercury column in order to measure accurately the amount of air pressure in the tank under test, and also to note the air leakage which will be indicated by a drop in pressure after the air supply has been shut off. A schedule of test pressures for the various compartments was compiled for this purpose. An additional piping lead extends through the firerooms and engine room with terminals conveniently located.

There is installed on the dynamo platform a motor driven air compressor for supplying air to the officers' and crew's galleys, the bakery and incinerator plant. In each of these rooms is located a gauge and a reducing valve to reduce the pressure to 20 pounds per inch for oil burning purposes. The compressor is adjusted to automatically maintain a pressure of 40 pounds on the system. As previously stated air can be obtained from the ship's system in case of the breakdown of the galley air compressor. Compressed air can also be obtained from the ordnance compressors through a suitable system of reducing valves.

Extensions from the ship's compressed air system are led to the 11 hygrostats located throughout the ship. The function of the hygrostat is to control the humidity of the air discharged from the thermo tanks of the ventilating sets by admitting steam through a perforated pipe to this air.

AIR COMPRESSORS.

Manufacturer	Westinghouse Air Brake Co.
Number	2
Steam cylinder, diameter, inches	11
Air cylinder, diameter, inches	11
Stroke, inches	12
Reservoir tanks, number	2
outside diameter, inches	24⅜
outside length, feet and inches	6– 6

GALLEY AIR COMPRESSOR.

Manufacturer	Allis-Chalmers Mfg. Co.
Capacity, cubic feet per minute	100
Motor, H. P.	20
Volts, D. C.	120

HEATING AND VENTILATING SYSTEM.

Steam radiators are installed in the emergency cabin on top of the bridge deck house; in the pilot and chart house, officers' quarters and optical shop on the bridge deck; in the officers' quarters, shops and sick bay on the main deck; and in the C. P. O.'s and crew's toilets on the second deck.

Steam is supplied from the auxiliary steam system through a reducing valve to a distributing manifold located on the port side of the engine room. Relief valve and gauge is provided at the manifold. There are nine circuits to supply steam to the radiators. The drains from these circuits are led to a manifold in the engine room which drains through a trap to a low pressure drain system leading to the auxiliary (*Dobbin* only) dynamo, and main condensers and the main feed and filter tank. A lead from the auxiliary exhaust system is provided to the supply manifold.

There are three circuits to supply steam to the eleven thermo-tanks located throughout the ship. Steam for humidifying the air discharged from the thermo-tanks is supplied from these circuits, the admission of the steam being regulated by hygrostats.

Throughout these systems expansion bends of the S type and U type are generously distributed and special care was taken to provide for expansion of the pipes in order to reduce to a minimum thermal strains on the pipe joints and insure tightness in service.

The living spaces, radio rooms and shops not listed above are heated by a ventilating system. The optical shop is also heated by a vent system in addition to steam radiators. Thermo-tanks are installed for heating air discharged from

the vent sets. Suitable bypass valves are provided on all thermo tanks in order that the vent system can be used to supply cold air when warm air is not required. Some of the vent sets are used for exhausting air from compartments and others for supplying air to compartments. Some of the sets which supply air to compartments are connected to thermo tanks as previously stated. The radio rooms are heated by electric heaters in addition to the ventilating system.

The data on ventilating sets is as follows, capacities in cubic feet per minute:

List of Ventilating Systems.

System No.	*Capacity*	*Cold Air*	*Warm Air*	*Exhaust Air*
1	2500	1200	1300	—
2	1600	—	—	1600
3	1600	1600	—	—
*4	3000	—	—	3000
5	1600	1600	—	—
6	2500	2500	—	—
7	600	—	—	600
8	2500	2500	—	—
9	200	—	200	—
10	200	—	—	200
11	2500	2500	—	—
12	5000	4000	1000	—
13	4000	4000	—	—
14	600	—	—	600
15	4000	4000	—	—
16	2500	—	2500	—
17	2500	—	2500	—
18	2500	2500	—	—
19	2500	—	2500	—

* Ventilation systems marked thus * are propeller type fans in W.T. boxes on superstructure deck.

System No.	*Capacity*	*Cold Air*	*Warm Air*	*Exhaust Air*
20	2500	—	2500	—
21	1600	—	—	1600
22	1600	—	—	1600
24	2500	—	—	2500
25	4000	4000	—	—
26	2500	2500	—	—
27	5000	—	5000	—
28	2500	—	2500	—
29	5000	—	5000	—
30	1600	—	—	1600
31	1600	—	—	1600
32	1600	1600	—	—
33	5000	—	5000	—
34	10,000	10,000	—	—
35	10,000	10,000	—	—
36	5000	5000	—	—
*37	3000	—	—	3000
*38	3000	—	—	3000
*39	3000	—	—	3000
40	2500	2500	—	—
42	600	600	—	—
44	1000	1000	—	—
45	200	—	—	200

AIR ESCAPES AND SOUNDING TUBES.

There is installed in all tanks into which liquids can be introduced a system of air escapes adequate in capacity to prevent the development of excessive pressure in the tanks when filling them. On the outlet of the fuel oil air escape pipes with a few exceptions is installed a deKrafft tank vent valve on the *Dobbin* and a ball float valve on the *Whitney* which permits the egress and ingress of air as the tank is filled and emptied. The

* Ventilation systems marked thus * are propeller type fans in W. T. boxes on superstructure deck.

deKrafft valve seals the tank to prevent the escape of oil vapors. Woven wire mesh is also installed around the discharge opening of the valve as a fire prevention measure. Sounding tubes are installed in many of the tanks, but in some cases access for sounding is obtained through a fitting installed in a manhole cover.

EVAPORATING AND DISTILLING PLANT.

The evaporating plant is located in the hold between Frames 121 and 126. In the *Whitney*, however, certain of the condensing equipment was installed above the dynamo platform level in a compartment above the evaporator room. A grating was installed in the evaporator room approximately half way between the second deck and the inner bottom. On the *Dobbin* all the pumps and fresh water manifolds were located for access from the floor plates, and the control of the evaporators is from the grating. On the *Whitney* some of the pumps were located accessible from the floor plates, others accessible from the grating, the operating features of the evaporators were accessible from the grating, and some of the operating details were installed in the compartment above the evaporators. The plant was designed for a capacity of 40,000 gallons of potable water per 24 hours.

The evaporating and distilling plant is of the low pressure type in accordance with the present practice of the Bureau of Engineering. The plants on the *Dobbin* and *Whitney* differ in very many respects, principally in that the plant on the *Dobbin* is designed for triple and on the *Whitney* for quadruple effect. Another difference is that the pumps for the evaporating plant of the *Dobbin* are steam operated and on the *Whitney* they are motor operated. The piping scheme for the *Dobbin* is shown diagrammatically in Fig. 2.

Source of Heat.

Heat is supplied from the auxiliary exhaust system and it is admitted to the first effect evaporator coils. In case the auxiliary exhaust supply is inadequate, auxiliary steam can be used,

Digitized by Google

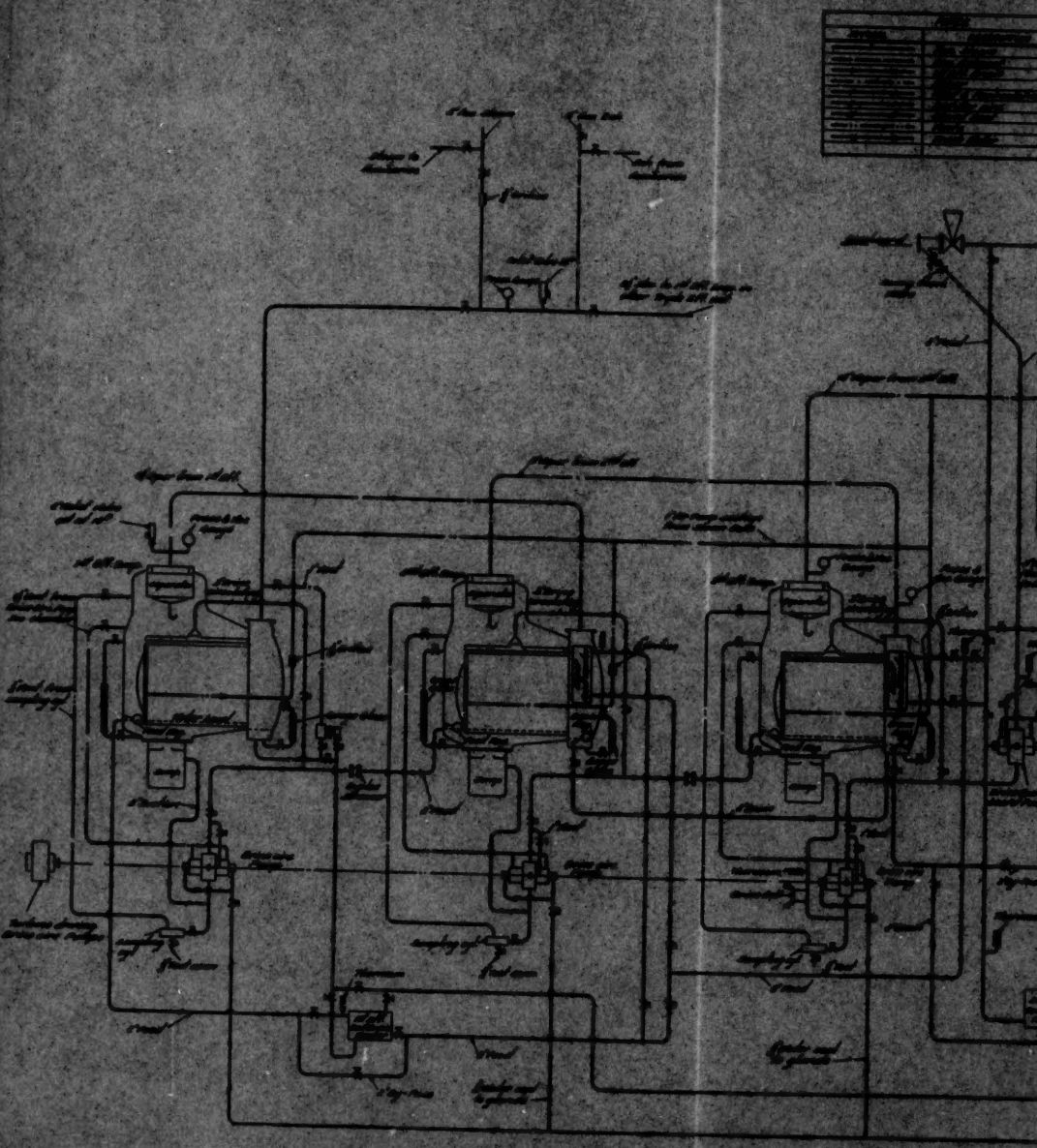
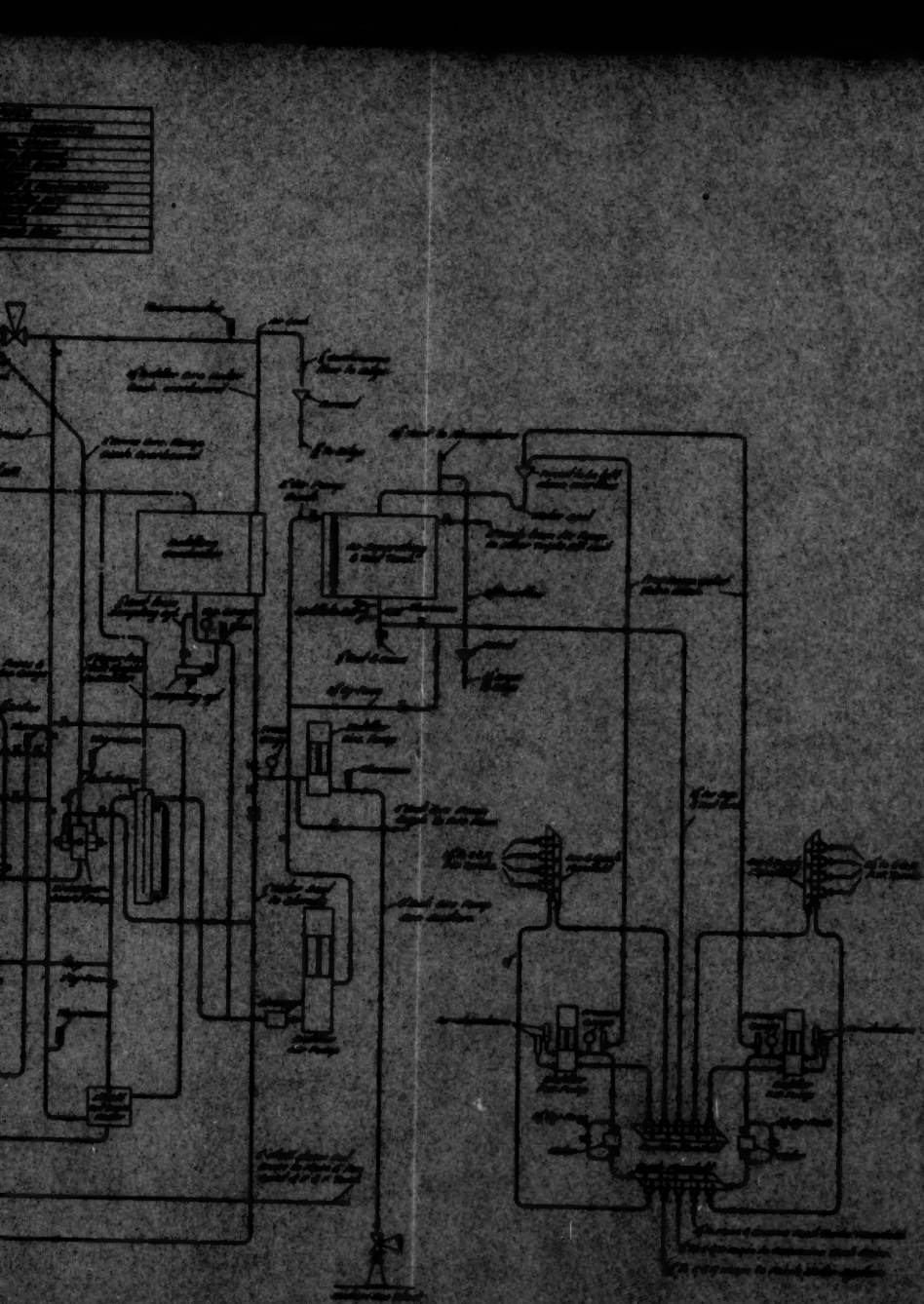


FIG. 2.—U. S. S. "DORSET." DIAGRAMMATIC ARRANGEMENT



ARRANGEMENT OF EVAPORATING PLANT PIPING.

an orifice valve being provided for this purpose. The condensate from the first effect coils is led, in the *Dobbin,* to two heaters, one for the port plant and one for the starboard plant. In the *Whitney* the drain from the first effect coils is led to one heater. From these heaters the drain is discharged to the auxiliary condenser (*Whitney* has no auxiliary condenser), dynamo condenser, main condenser or main feed tank. This constitutes the only heat supply for the operation of the plant. The vapor from the first effect evaporator is led to the coils of the second effect evaporator. The vapor from the third effect evaporator is led, in the case of the *Dobbin,* to the distilling condenser, but in the case of the *Whitney,* to the coils of a fourth effect evaporator, and the vapor from the fourth effect evaporator is led to the distilling condenser. Feed heaters are installed in the coils of the second and third effect evaporators on the *Dobbin* and in the coils of the second, third and fourth effect evaporators of the *Whitney.* The condensate from the second effect evaporator coils is led to the coils of the third effect evaporator, thence to a third effect coil drain heater, and to the air pump suction in the *Dobbin.* In the *Whitney* the condensate of the second effect evaporator coils is led to the coils of the third effect, thence to the coils of the fourth effect, thence to the fourth effect coil drain heater and to the condensate pump suction. In the *Dobbin* two third effect coil drain heaters are installed; on the *Whitney,* one fourth effect coil drain heater is installed for the port and starboard plants. The drains are controlled by means of drain regulators for each evaporator.

Condensing Equipment.

The distilling condenser receives the vapor from the last effect evaporator. On the *Dobbin* the condensate and non-condensible gases are drawn from the condenser by means of an air pump which discharges to an air separating tank, one tank being provided for both port and starboard plants. On the *Whitney* a centrifugal pump is used to draw the

condensate from the condenser and discharges to one air separating tank similar to the plant of the *Dobbin*, and the non-condensible vapors are removed from the condenser by means of a radojet manufactured by C. H. Wheeler Mfg. Co. The condenser circulating pumps in addition to providing condenser circulating water discharge to the firemain.

Venting of Coils.

On account of the air present in the auxiliary exhaust and also in the vapor from the evaporators a system of venting coils was installed which is connected to the condenser system. The discharge from the several coils is through an orifice which regulates the amount of air discharged.

Brine Recirculating System.

Centrifugal pumps are employed for recirculating the brine. The brine is drawn from sump tanks so arranged that they will retain the scale from the evaporators and at the same time prevent its admission to the pump suctions. The brine is discharged to the evaporator from which it is drawn and sprayed over the coils and a portion discharged into the next effect evaporator. The circulating pump of the last effect evaporator discharges to the suction of the brine overboard discharge pump.

Feed System.

Feed is obtained from the circulating water overboard discharge from the condenser. In the *Dobbin* this is led to the third effect coil drain heater, to the third effect coil heater, to the second effect coil heater, to the first effect coil drain heater, and to the first effect evaporator. In the *Whitney* the feed is led to the radojet condenser, to the fourth effect coil drain heater, to the first effect coil drain heater, to the fourth effect coil heater, to the third effect coil heater, to the second effect coil heater, and to the first effect evaporator.

Gland Seals and Pump Vent System.

In order to maintain an efficient vacuum the glands of the brine circulating pumps and brine overboard discharge pumps are provided with water from the circulating system for sealing purposes.

In order to efficiently vent the pumps and prevent their becoming air bound and failing to operate, air vents are provided on the suction chambers and on the discharge chamber of the pumps. The vent was led to the evaporator directly above the pump. It was considered essential that air be vented from the system in order to have a solid column of water from the evaporator back to the evaporator.

Salinity Indicating System.

A salinity indicator system, manufactured by the Leeds & Northrup Co., is installed to indicate the salinity of the brine being circulated through the last effect evaporator in order to determine the rate of discharge overboard of the brine and also to indicate the salinity of the water discharged from the air separating tank on its way to the ship's system.

Fresh Water Pumping System.

Two fresh water pumps are provided and by means of a manifold and piping system one pump can be used to draw water from any tank and discharge to any other tank, or to the ship's system, and the other pump to draw from the air separating tank and discharge to any tank or the ship's system as occasion might require.

EVAPORATOR DATA.

	Dobbin	*Whitney*
Manufacturer.....Navy Yard	Phila.	Boston
Number	6	8
Heating surface, each, square feet.....	210	245

	Dobbin	*Whitney*
Tubes:		
Number, each	157	156
Diameter outside, inches	1½	1½
Thickness, mils	65	72
Length between tube sheets, feet and inches	3– 4⅜	3–11 13/16
Openings in shell:		
Steam inlet, first effect, diameter, inches	5½	6
second effect, diameter, inches	5½	9
third effect, diameter, inches	8	9
fourth effect, diameter, inches	—	9
Vapor outlet, first effect, diameter, inches	5½	9
second effect, diameter, inches	8	9
third effect, diameter, inches	15	9
fourth effect, diameter, inches	—	14
Coil drain outlet, first, second and third effect, diameter, inches	2	1½
fourth effect, dia., inches	—	2½
Feed inlet, diameter, inches	2	1½
Vent, coil, diameter, inches	1	¾
brine circ. pump suct. chamber, diameter, inches	1¼	½
Gauge glasses, head and shell, diameter, inches	¾	1
Brine circulating pump discharge, dia., inches	3	2½
suction from sump., diameter, inches	5	5

Float operated regulator in shells for automatic control of feed.

External float operated regulator for first effect, float operated regulator in coil heads for others.

FIRST EFFECT COIL DRAIN FEED HEATER DATA.

	Dobbin	*Whitney*
Type	4 pass straight tube	2 pass U-tube
Number	2	1
Heating surface, each, square feet	51.5	45
Tubes:		
Number, each	180	98
Diameter, outside, inch	⅝	⅝
Thickness, mils	49	49
Length, between tube sheets, inches	21	..
Connections:		
Coil drain inlet, diameter, inches	2	2
outlet, diameter, inches	2	2
Feed, inlet, diameter, inches	2	3
outlet, diameter, inches	2	3

COIL DRAIN FEED HEATER DATA.

	Dobbin	*Whitney*
Effect	third	fourth
Type	single pass straight tube	2 pass U-tube
Number	2	1
Heating surface, each, square feet	83	45
Tubes:		
Number, each	109	98
Diameter, outside, inch	⅝	⅝
Thickness, mils	65	49
Length between tube sheets, feet and inches	4– 8	..
Connections:		
Coil drain inlet, diameter, inches	2	3
outlet, diameter, inches	2	3
Feed inlet, diameter, inches	2	3
outlet, diameter, inches	2	3

DISTILLER DATA.

	Dobbin	*Whitney*
Type: cylindrical, horizontal, two pass, tubes	straight	curved
Number	2	2
Cooling surface each, square feet	552	450
Tubes:		
Number, each	494	607
Diameter, outside, inch	⅝	⅝
Thickness, mils	65	65
Length between tube sheets, feet and inches	6–10	4–8⅞ mean
Connections:		
Vapor inlet, diameter, inches	15	14
Air pump suction, diameter, inches	5½	..
Circulating water inlet, diameter, inches	5½	5
outlet, diameter, inches	5½	5
Air cock on water chest, diameter, inch	½	⅜
Condensate pump suction, diameter, inches	..	2½
Radojet pump suction, diameter, inches	..	(2)–1½

VAPOR FEED HEATER DATA.

	Dobbin	*Whitney*
Effect	2 & 3	2, 3, & 4
Type: tubes	straight	U-tube
Number	4	6
Heating surface each, square feet	51.5	57.0

Tubes:		
Number, each	180	130
Diameter, outside, inch	⅝	⅝
Thickness, mils	49	65
Length between tube sheets, inches	21	..
Connections:		
Feed inlet, diameter, inches	2	2
outlet, diameter, inches	2	2

FRESH WATER METER DATA.

	Dobbin	*Whitney*
Type	Worthington turbine meters	
Number	2	2
Capacity per minute, each, gallons (125 degrees F.)	40	41
Connections:		
inlet, diameter, inches....3 inches reduced to	2½	2
outlet, diameter, inches....3 inches reduced to	2½	2

REFRIGERATING PLANT.

The ice machines and cold storage rooms are located on the third deck and extend from Frame 65 to 98. The cold storage rooms, in addition to providing for the needs of the ship, are sufficient in number and capacity for cargo purposes and can supply all reasonable service demands for refrigerated provisions. Two of the rooms are for cargo meats and three for the ship's service, consisting of a meat room, a butter and egg room, and an officers' locker room.

Brine circulation is used throughout the rooms. In the large rooms radiators are installed on two or more circuits with a view to reducing the time the brine would be in a room to the minimum, compared with a single circuit installation. Such an arrangement would discharge brine from the rooms at a lower temperature than a single circuit system and thus improve the efficiency of the installation. Furthermore, there may be times when only one radiator is required.

Each of the two CO_2 compressors is of sufficient capacity to operate the plant under normal conditions and the gas piping system was laid out with this in view. The gas is discharged to the CO_2 condenser through an oil trap (*Dobbin* has two,

EVAPORATING PLANT.

RECIPROCATING PUMPS ("DOBBIN" ONLY).

Use	No.	Cylinder Diameter		Connections				Stroke inches	Maker
		Steam inches	Water inches	Steam inches	Exhaust inches	Suction inches	Disch'rge inches		
Distiller Circulating.....	2	12	14	2	2½	8	7	16	Dow Pump & Diesel Engine Co.
Distiller Air.....	2	7½	2–14	1	1½	6	5	10	Worthington Pump and Machinery Corporation.
Distiller Fresh Water.....	2	4½	6	½	¾	3	2½	6	Worthington Pump and Machinery Corporation.

CENTRIFUGAL PUMPS TURBINE DRIVEN ("DOBBIN" ONLY).

Use	No.	Capacity G. P. M.	Discharge Head feet	Suction inches	Discharge inches	R. P. M.	Turbines				Maker
							No.	H.P.	Steam inches	Exh. inches	
Evap. Brine Circ.*.....	6	200	16	5	3	1800	2	10	2	4	Buffalo Steam Pump Co. Standard Turbine Corp.
Evap. Brine Overboard.....	2	50	72	2½	2	2000	2	5	2	4	Buffalo Steam Pump Co. Standard Turbine Corp.

*Three pumps were direct connected through flexible couplings to one turbine.

EVAPORATING PLANT.

CENTRIFUGAL PUMPS MOTOR DRIVEN ("WHITNEY" ONLY).

Use	No.	Capacity G. P. M.	Discharge Head feet	Suction inches	Discharge inches	R. P. M.	Motor H. P.	Maker
Evaporating Brine Overboard..	2	40	70	2½	2	1700	4	Buffalo Steam Pump Co. General Electric Co.
Distiller Circulating.....	2	800	40	6	5	1700	15	
Distiller Fresh Water.....	2	50	72	2½	2½	1700	4	
Evaporating Brine Circulating*.....	8	200	15	5	3	1200	10	
Condensate	2	50	40	2½	2½	1700	2	

*Four pumps were direct connected through flexible couplings to one motor.

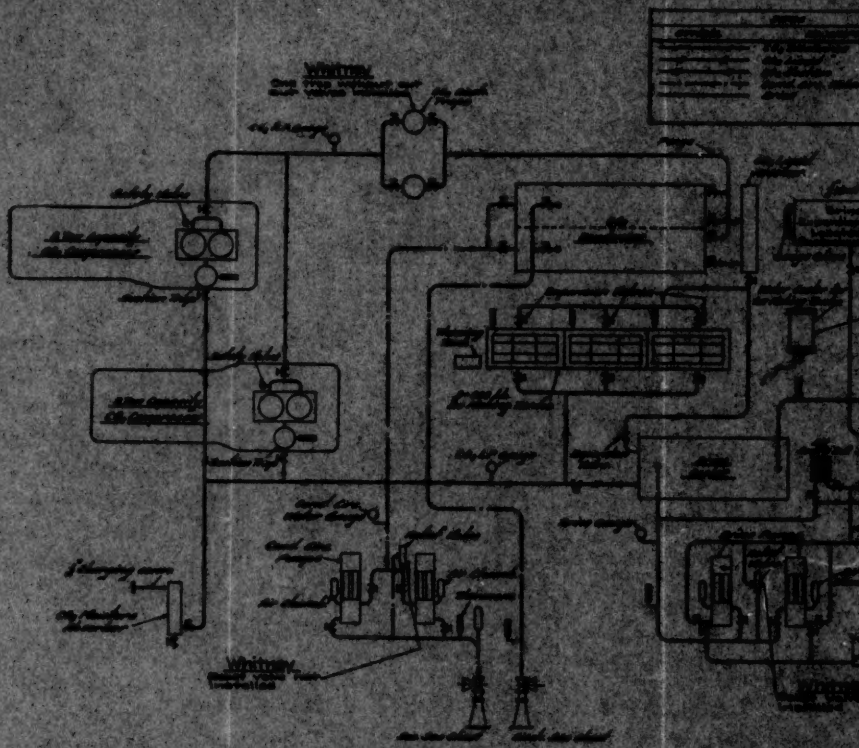
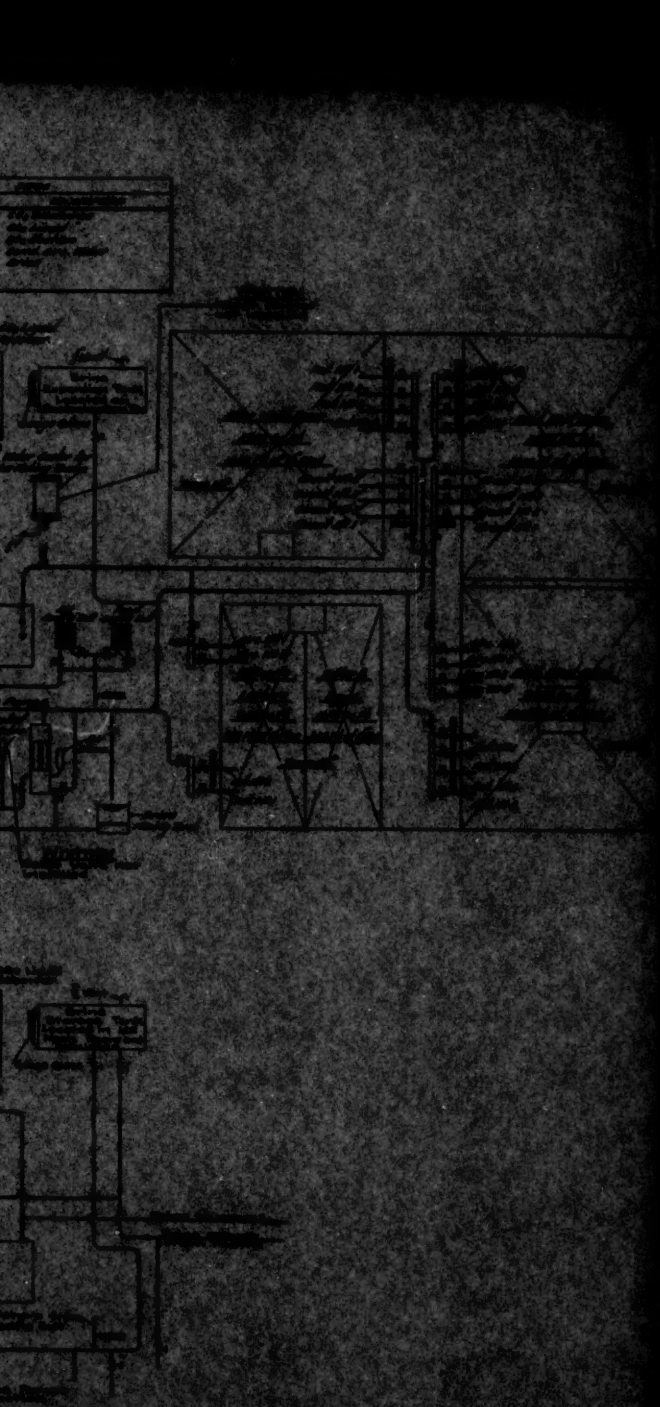


FIG. 3.—"DOBBIN" AND "WHITNEY" DIAGRAMMATIC ARRANGEMENT



Whitney has one). The condenser is of the rectangular shell type divided into two compartments. Each compartment is of sufficient capacity to operate one compressor. The liquid CO_2 is piped to an expansion valve on the brine cooler (*Dobbin* piped also to ice-making tanks, *Whitney* not so piped). The gas is returned to the compressors through scale traps located on the compressor cylinders. The circulating pumps for the condenser draw from a sea valve located below in the fireroom and each pump is of ample capacity to operate one compartment of the CO_2 condenser, or for operation with one compressor. In case of emergency, however, circulating water can be supplied from the flushing system. For diagrammatic arrangement of piping see Fig. 3.

The brine circulating pumps, each of sufficient capacity for operation with one compressor, discharge to the two scuttle butts and to the brine cooler. The brine from the cooler is discharged to the cold storage rooms. (In the case of the *Whitney* the brine is also discharged to the ice making tanks; on the *Dobbin* CO_2 is used for ice making tanks.) The brine is returned to the ice machine room; in the *Whitney,* to a storage tank located in the ice machine room; in the *Dobbin,* to the brine circulating pump suction and a pipe was run from the return to an expansion tank located above on the superstructure deck. It was unnecessary to lag this expansion tank and only light lagging was required on the riser pipe. In the *Whitney* the brine is drawn from the storage tank by the brine circulating pumps and discharged as previously described. A suitable gauge glass was installed on the expansion tank and the storage tank. A lead was run from the return line to the mixing tank so that the liquid can be recirculated through the system while it is being charged with calcium chloride, and in this way facilitate the charging of the system. The usual vents and drain valves are installed in each of the radiators in the cold storage rooms. Vents were placed at all high places in the system, including a purge valve for the CO_2. The pumps were fitted with air chambers. Pump relief valves

were installed on the *Dobbin,* but none on the *Whitney.* The compressors are fitted with a forced lubrication system and the engine with a sight feed lubricating system. A water cooler was installed on the *Dobbin* but not on the *Whitney,* for cooling water required to fill the icemaking cans. (The object was to reduce the amount of refrigeration required from the icemaking tanks to a minimum.) Return brine is used as a cooling agent.

A moisture absorber or gas dryer was installed through which CO_2 is to be charged into the system. The pipe from the dryer is led to the suction of the compressors. The dryer was so designed that it can be readily disassembled for the renewal of calcium chloride. Each compressor is operated by a reciprocating steam engine. The return gas is led to the compressor crank case, the ice machines being of the enclosed crank case type as manufactured by the York Manufacturing Co. The CO_2 condenser and the brine cooler were also manufactured by the York Co. The circulating pumps were manufactured by the Worthington Pump & Machinery Corporation. The usual thermometers, gauges and log desk were installed, also a suitable work bench.

In order to charge the system about 750 pounds of liquid CO_2 and about 6000 pounds of calcium chloride were required.

CO_2 Compressors.

8-Ton Compressors, number	2
Steam cylinder, diameter, inches	8
stroke, inches	8
Nozzle, steam, diameter, inches, tap	1½
exhaust, diameter, inches, tap	2
Compressor, cylinders, number	2
diameter, inches	4¼
stroke, inches	5
Combined capacity refrigerations, tons per 24 hours	16

Icemaking Tank.

Tanks, number	3
Length of 1-inch extra heavy pipe, feet	133
Number of icemaking cans per tank	9
Dimensions of ice can, inches	2⅜ × 12 × 22⅝

Ice cake, weight, pounds	17
Pounds per 24 hours	600
Total combined capacity ice per 24 hours, pounds	1800

Cold Storage Rooms.

Meat Room, starboard	Frames 65 to 74
port	Frames 65 to 74
Volume, cubic feet	2850
Radiator circuits, number	4
Length, 1¼-inch galvanized pipe, feet	1300
Ratio V to P	2.2
Designed temperature, degrees F.	15
Meat room, port	Frames 75 to 82
Volume, cubic feet	2700
Volume, vestibule, cubic feet	300
Length of 1¼-inch galvanized pipe, meat room	1249
vestibule	82
Ratio V to P, meat room	2.2
vestibule	3.6
Designed temperature, degrees F., meat room	15
vestibule	40
Officers locker and butter and egg rooms, port	Frames 84 to 90
Volume, cubic feet,	
officers lockers	920
butter and egg room	1125
vestibule	300
Radiator circuits, number,	
officers lockers	1
butter and egg room and vestibule	1
Length of 1¼-inch galvanized pipe, feet,	
officers lockers	260
butter and egg room	330
vestibule	98
Ratio V to P,	
officers lockers	3.5
butter and egg room	3.4
vestibule	3.0
Designed temperature, degrees F.,	
officers lockers	40
butter and egg room	40
vestibule	40

Condenser Circulating Pump.

Pumps, number	2
Cylinder, steam, diameter, inches	6½
water, diameter, inches	7
stroke, inches	8

Nozzle, steam, diameter, inch, tap	¾
exhaust, diameter, inch, tap	1
suction, diameter, inches	3½
discharge, diameter, inches	3

Brine Circulating Pump.

Pumps, number	2
Cylinder, steam, diameter, inches	6½
water, diameter, inches	8
stroke, diameter, inches	8
Nozzle, steam, diameter, inch, tap	¾
exhaust, diameter, inch, tap	1
suction, diameter, inches	4
discharge, diameter, inches	4

CO_2 Condenser.

Nozzle, circulating water inlet, inches	3
outlet, inches	3
Drain, diameter, inches	2
Compartments, independent, number	2
Gas inlet chamber inches	1¼
Liquid receiver, diameter, inches	6
length, feet	3
Liquid CO_2 outlet, diameter, inch	¾
Length of 1-inch extra heavy pipe, each compartment, feet	830
Length in feet and inches	7– 0
Width in feet and inches	3– 3
Height in feet and inches	6– 4½

Brine Cooler.

Outside diameter, feet and inches	3– 6
Length, feet and inches	10– 0
Brine nozzle inlet, diameter, inches	4
outlet, diameter, inches	4
Liquid CO_2 inlet, diameter, inch	½
Gas outlet, diameter, inches	1¼

Brine Storage Tank.

Tank, number	1
Outside diameter, feet and inches	2– 6
Length, feet and inches	7– 0
Brine, inlet, diameter, inches	1¼
Air outlet, diameter, inch	½

Brine Mixing Tank.

Tank, number	1
Inside diameter, feet	2
Length, feet	3
Suction nozzle, diameter, inches	2

ELECTRICAL INSTALLATION.

The turbo-generators are located on the dynamo platform in the engine room, two on the starboard and one on the port side. The units on the starboard side are of 200 K.W. capacity and the other of 100 K.W. capacity. The governors of the turbines are fitted to operate for overspeed, for excessive back pressure, and for tripping the circuit breakers.

DATA.

100 K.W. and 200 K.W. Units.

Type	General Electric Co., M.P.C. interpole, horizontal, direct current
Voltage	125
R. P. M., generator	1200
Winding	Compound
Lubrication	Forced

	Dobbin	*Whitney*
Turbine manufacturer	Moore Steam Turbine Corp.	General Electric Company
Steam pressure, pounds	200	200
Steam inlet, diameter, inches	3	3 —100 K.W. set 3½—200 K.W. set
Exhaust outlet, diameter, inches	14	14
Reduction gear, R. P. M.	4500 to 1200	4500 to 1200

MAIN SWITCHBOARD.

The main switchboard was designed and built at the Philadelphia Navy Yard for the *Dobbin*. It is located on the dynamo platform forward amidships. The board is divided into 8 sections arranged as follows:

Generators	3 sections
Power	3 sections
Lighting	1 section
Searchlights	1 section

The generator sections are fitted with the usual instruments such as voltmeters, ammeters, watt-hour meters, main and equalizer switches, together with a single pole circuit breaker for each generator, control handles for shunt field rheostats and so on. Each circuit breaker opens automatically when the governor of its turbo-generator trips and closes its throttle valve.

The power, lighting and searchlight sections are fitted with double pole circuit breakers 5 and 12 point voltmeter switches, voltmeters, ammeters, etc., and supply power to the several auxiliary power and lighting distribution panels, searchlights and auxiliary apparatus not supplied through auxiliary distribution panels.

The interior communication switchboard consists of one section and is located on the port side of the main switchboard. It supplies all the 20 and 125 volt interior communication systems, together with two interior communication 125 volt to 20 volt motor generators. The usual voltmeters, ammeters, ground detector instruments, and so on are mounted on the panel.

POWER SYSTEM.

The main switchboard distributes power throughout the ship over 25 circuits supplying the following appliances and 16 auxiliary power distributing panels:

Circuit No.	*Supplies.*
P 100	Radio apparatus, bridge deck
P 102	Vent motor No. 34—10,000 cu. ft., engine rm., Fr. 112P
P 103	Vent motor No. 35—10,000 cu. ft., engine rm., Fr. 112S
P 104	Air compressor No. 1, torpedo shop, 3rd dk., Fr. 63S
P 105	Air compressor No. 2, torpedo shop, 3rd dk., Fr. 63P
P 106	Aux. panel No. 3, second dk., Fr. 56S
	Aux. panel No. 2, second dk., Fr. 36P
P 107	Aux. panel No. 8, second dk., Fr. 91P
	Aux. panel No. 12, main dk., Fr. 72S
P 108	Aux. panel No. 9, second dk., Fr. 123S
	Aux. panel No. 11, second dk., Fr. 144S
P 109	Torpedo hoist, second dk., Fr. 70P

P 110—Aux. panel No. 5, second dk., Fr. 57S
P 111—Aux. panel No. 4, second dk., Fr. 57P
P 112—Aux. panel No. 7, second dk., Fr. 82S
P 113—Aux. panel No. 6, second dk., Fr. 82P
P 114—Aux. panel No. 10, second dk., Fr. 111P
P 115—Aux. panel No. 13, main dk., Fr. 100S
P 116—Aux. panel No. 14, main dk., Fr. 108P
P 117—Aux. panel No. 15, main dk., Fr. 119P
P 118—Aux. panel No. 16, bridge dk., Fr. 74C
P 119—Battery charging panel, main dk., Fr. 114S
P 120—Portable electric welder, second dk., Fr. 83S
P 121—Int. comm. switchboard, dynamo flat
P 122—Aux. panel No. 1, hold, Fr. 106S
P 123—Electric shop, bake ovens, second dk., Fr. 83P&S
P 124—Galley air compressor, dynamo flat
P 125—Evaporating plant pumps, Aux. panel No. 17 (*Whitney* only)
P 130—Shore connection and destroyer supply, second dk., Fr. 82–86

The auxiliary power distribution panels were designed in several sizes containing from 6 to 18 circuits, and supply current to the motors in their vicinity for operating machine tools, ventilating sets, laundry machinery, galley appliances, etc.

Electrical Shore Connection and Destroyer Supply Circuits.

There is one shore connection box having a capacity of 3600 ampères located on the port side at Frame 86 accessible from the main deck. This connection can be used for either supplying power to or receiving power from an outside source. There are four destroyer terminal boxes provided having a capacity of 400 ampères each, located 2 port and 2 starboard, near Frame 82, accessible from the main deck. By means of portable connections from these boxes 4 destroyers can be supplied with electric current at one time.

Battery Charging.

A storage battery charging room equipped with charging panel, racks for batteries, etc., is arranged for charging and repairing auxiliary lighting, gun firing, and other small batteries.

Motors.

There are 101 motors on the *Dobbin* and 111 on the *Whitney* varying in size from ¼ to 120 H.P., installed at various places throughout the ship, used for the operation of machine tools, laundry machinery, etc.

Ventilating Motors.

There are 42 motors varying from ⅛ to 7.5 H.P. used for operating ventilating systems located throughout the ship. There is a total of 87 portable bracket fans installed throughout the living quarters.

Electric Heaters.

Outlets for electric heaters are provided in the radio room, gyro store room, pilot house and bridge. Nine electric heaters were furnished—two of 500 watts; two of 500 watts, non-magnetic; three of 1000 watts; two of 3000 watts capacity. Four disk heaters were supplied for the sick bay.

Lighting System.

Extending from the main switchboard are 9 battle lighting circuits, 7 general lighting circuits, and 5 searchlight circuits. The battle and general lighting circuits supply power to 7 lighting distribution panels located as follows:

Circuit No.	*Supplies.*
BL 1	Dist. Panel No. 1, bridge dk., Fr. 72
L 15	Dist. Panel No. 1, bridge dk., Fr. 72
BL 2	Dist. Panel No. 2, superst. dk., Fr. 72
L 16	Dist. Panel No. 2, superst. dk., Fr. 72
BL 3	Dist. Panel No. 3, main dk., Fr. 72
L 17	Dist. Panel No. 3, main dk., Fr. 72
BL 4	Dist. Panel No. 4, second dk., Fr. 128
L 18	Dist. Panel No. 4, second dk., Fr. 128
BL 5	Dist. Panel No. 5, second dk., Fr. 36
L 19	Dist. Panel No. 5, second dk., Fr. 36
BL 6	Dist. Panel No. 6, second dk., Fr. 90
L 20	Dist. Panel No. 6, second dk., Fr. 90

BL 7—Dist. Panel No. 7, second dk., Fr. 122
L 21—Dist. Panel No. 7, second dk., Fr. 122
BL 8—Engine room, fire room and dynamo flat
BL 9—Engine room, fire room and dynamo flat
BL 10—24-inch Searchlight No. 1
BL 11—24-inch Searchlight No. 3
BL 12—24-inch Searchlight No. 2
BL 13—24-inch Searchlight No. 4
BL 14—12-inch Signal searchlight No. 5

The distribution panels were designed in several sizes and the circuits from them supply the general and battle lights in their vicinity.

Auxiliary Lighting.

The auxiliary lighting system consists of two independent circuits, one for supplying lighting through relays to several compartments on the second deck and above, the other circuit for lights below the second deck. Each circuit is energized by a separate 200 ampère-hour storage battery. The system operates on 30 volts, and the relays are so arranged that they will cut in the auxiliary lighting system when the general lighting system fails.

Searchlights.

There are four General Electric Co. high power 24-inch searchlights, each fitted for remote mechanical control below the searchlight. Two are installed forward above the superstructure deck at Frame 60, and two aft at Frame 148. There is one 12-inch portable arc searchlight with signal shutter provided, and outlets located for it at the port and starboard end of the bridge.

Signals.

Two truck lights, one on each mast; two blinker lights mounted on the port and starboard end of the yard arm on foremast; a man-overboard and breakdown light on the main-

mast; man-of-war and absentee lights on the mainmast; screen speed, flag, range, anchor, towing, stern and masthead lights are provided.

Gyro Compass System.

There is a master compass located on the third deck near Frame 50 with the switchboard and storage batteries in a locked enclosure. The compass installed on the *Dobbin* was manufactured by the Sperry Gyroscope Company and that on the *Whitney* by the Arma Engineering Company, Inc. On the *Dobbin* to operate this system two motor generators were provided and they were located in the locked enclosure, on the *Whitney* one turbo generator is provided and it is located on the starboard side of the dynamo platform in the engine room. The system operates repeaters, one at each pelorus, one at the steering stand on the bridge, one in the radio compass room, and one in the steering engine room. The one in the steering engine room can be located near the control wheel of the steering engine or near the hand steering wheels.

Interior Communication System.

The interior communication system circuits are energized from the interior communication switchboard, and consist of the systems listed below:

Call bells for voice tubes, and pantry.
Thermostat for fire alarm.
General alarm gongs.
Boat hour gongs.
Shaft revolution indicator.
Steering telegraph.
Engine order telegraph.
Loud speaking telephones.
Ship's service telephones, automatic.
Rudder angle indicator.
Salvo bells.

Fuel oil tank indicators.
Gyro compass.
Salinity indicator.
Low oil pressure alarm.
Torpedo firing and ready lights.
Pneumercator cleavage gauge.

The telephone system installed is of the automatic type, the switchboard being located at Frame 93, second deck (center line).

Sound and Radio Installation.

The compensator or sound receiver for the sound system is installed in the radio room. The microphones for this system are located in a tank forward, one system port and one starboard. The usual inertia plates and sound screens are installed. This system is installed to locate direction of sources of sounds from passing ships, light ships, and also for determining depth of water or soundings.

The radio installation consists of one 2-K.W. spark transmitter, one radio telephone transmitter and receiver, one long wave receiver, one medium wave receiver, and a radio compass. This equipment is located in the radio room on the bridge deck. The space is divided into a transmitting and receiving room, a radio compass room, a store room, and a storage battery room. In the store room is located the motor generator. Facilities for charging the storage batteries and venting the compartment are provided. The bulkheads, floor and ceiling of the receiving-transmitting room are sound proofed.

Lighting is provided from the ship's system with an auxiliary system supplied from a storage battery which automatically goes into service if the lighting system should fail. Interior communication is provided by the ship's telephone system and voice tubes.

The main radio transmitter is a 2-K.W., 500 cycle quenched spark set with wave changer for nine wave lengths. The motor for this system operates at 120 V., D. C. and generates

A. C. at 250 V., 500 cycles for the transmitter. This transmitter is for radio communication between the ship and other ships or shore stations at a distance. A short range telephone transmitting receiving set is installed by means of which the radio operator or the commanding officer on the bridge may converse with or telegraph to a receiving station not over ten miles away. The radio receiving equipment consists of two receivers complete with vacuum tube detector, amplifiers, filament and plate storage batteries. One receiver is designed for medium wave reception and the other for long wave reception.

A model "T" radio compass is installed which consists of a special loop antenna placed in a wooden deck house, secured to the deck above the radio compass room. A shaft from the compass extends into the compass room to which a handwheel is secured for controlling the position of the compass in azimuth. The receiver and amplifier are located in this room. This apparatus is used for determining the direction of sending stations either on ship or on shore. For convenience there is located in this room a Sperry gyro compass repeater to assist the operator in determining bearing of the station received.

The main antenna is a 6-wire, flat top, "T" type, 19 feet wide, 260 feet between spreaders, and is suspended by a bridle and 4 safety strain insulators, from the fore and the mainmasts. It is approximately 160 feet above the waterline. From the center spreader the rattail leads down to the insulator in the deck to the radio room. Suitable protection for the ship's personnel is provided on the deck where the rattail passes through. The antenna is used for the main transmitter and safeguards are provided to prevent energizing it when the radio telephone antenna is connected to the radio telephone equipment.

The radio telephone antenna is a 6-inch diameter 6-wire cage, 60 feet long, and is suspended by two safety strain insulators from an insulated stay between the fore and the mainmasts. The top of the antenna is approximately 110 feet above the water line. The antenna slopes at an inclination of 30 degrees aft from the vertical and is direct connected to the radio tele-

phone insulator on the after side of the radio house. Suitable protection for the ship's personnel is provided around this lead.

BOILERS.

There are two boilers of the Bureau of Engineering express type designed for burning fuel oil. Each boiler is located in a watertight compartment. The smokepipe is divided into two compartments, one for each boiler. Two forced draft blowers are provided per fireroom. There are on the boilers three blower burners of the Babcock & Wilcox Lodi type. A small hand pump is provided for starting the burners if occasion requires. The blower burners are of ample capacity for normal operating conditions, but in emergency the forced draft blowers can be used and the rate of combustion increased.

BOILER DATA.

Manufacturer	Navy Yard
Number of boilers	2
Designed working pressure, pounds per square inch	215
Steam drums, per boiler	1
Diameter of steam drums inside, inches	48
Water drums, per boiler	2
Diameter of water drums inside, inches	24
Length of steam drums inside, feet and inches	13– 3 5/16
water drums inside, feet and inches	12– 4 3/4
Thickness of steam drums, inch	19/32
steam drums, tube sheets, inches	1 29/64
water drums, inch	17/32
water drums, tube sheets, inch	7/8
Number and size of manholes in steam drum, inches	1—11 by 15
Number and size of manholes in each water drum, inches	2—11 by 15
Number rows generating tubes (2 groups)	4
Number generating tubes, per boiler	174
Number rows circulating tubes (2 groups)	34
Number circulating tubes, per boiler	2482
Outside diameter generating tubes, inches	1 3/8
circulating tubes, inches	1 1/8
Thickness generating tubes, mils	134
circulating tubes, mils	120
Furnace volume, per boiler, cubic feet	560
Heating surface, per boiler, square feet	7000
Safety valves	manufactured by Consolidated Safety Valve Co.

Four safety valves, duplex, inches	4½
Number registers, per boiler	3
Type register	Babcock & Wilcox "Lodi"
Atomizers	Babcock & Wilcox "Lodi"
Number smoke pipes	1
Diameter smoke pipe, feet and inches	7– 2
Height smoke pipe above furnace floor, feet, inches	84– 0½

FORCED DRAFT BLOWER DATA.

Number per fire room	2
Manufacturer,	
Terry Steam Turbine Co.	turbine
B. F. Sturtevant Co.	blower
Fan diameter, outside, inches	40
Width of tip opening, inches	7
Nozzle, steam, diameter, inches	2½
exhaust, diameter, inches	4
Back pressure allowable, pounds per square inch	10
Capacity air at 5 inches water pressure, cubic feet per minute	25,000

MAIN AND AUXILIARY STEAM.

The main steam line is led from each boiler to a Y fitting secured to the maneuvering valve. Steam is supplied from each boiler to the auxiliary steam system which loops around the fire rooms and engine room. Manifold castings are provided from which steam is supplied to the pumps, ice machine room, forward deck machinery, whistle, forced draft blowers, for steaming out sea chests, for foam fire extinguishers, for tube blowing, for fuel oil tank heating system, and so on, in the fire rooms. In the engine room a similar arrangement in principle exists, steam being supplied to pumps, turbo generator sets, vacuum augmenter, heating system, galley system, laundry, evaporator room, after deck machinery, to sea chests, and so on. The pipes were bent to avoid the use of expansion joints.

AUXILIARY EXHAUST SYSTEM.

The auxiliary exhaust system extends through the engine and fire rooms and receives the exhaust from auxiliary machinery outside of machinery spaces. An exhaust line leads to the main condenser, dynamo condenser, auxiliary condenser

(no auxiliary condenser was installed on the *Whitney*), evaporator room, the low pressure turbine through a spring loaded back pressure valve, and to the atmosphere. The turbo generators exhaust directly to the dynamo condenser, but the exhaust can be led to the auxiliary condenser (on *Dobbin* only), to main condenser, and to the atmosphere. Back pressure valves are provided on the condensers through which the exhaust must pass.

STEAM DRAINAGE SYSTEM.

An adequate system of steam drainage is provided for the drainage of condensate from auxiliary machinery and from low points in the piping systems. The auxiliary machinery cylinders in the engine and fire rooms drain to funnels which in turn drain to the inner bottom reserve feed tanks, thus providing a visible indication of whether or not the drain valves on auxiliary machinery are leaking and so on. The drains from the heavy pressure system are in every instance from traps which in turn discharge to the feed and filter tank, as well as to the condensers.

SUCTION AND DISCHARGE SYSTEM.

In the engine and fire rooms are located the pumps required for the operation of the main propelling machinery. In the forward fire room are located one cargo fuel oil pump, two fuel oil service pumps, one fuel oil supply pump, one auxiliary feed pump, one fire and bilge pump. In the after fire room are located pumps for the same services except the fuel oil supply pump. In the engine room are located two main feed pumps, one fire and bilge pump, one flushing pump. The other pumps have been described in connection with other systems.

The usual provisions for pumping out the main drain and local bilges are installed in the engine and fire rooms. The main feed pumps discharge through the main feed heater, which may be by-passed, the heating agent being auxiliary exhaust.

The fuel oil system is so arranged by means of manifolds and pipes that the oil can be drawn from any tank and discharged to any tank with certain limitations, or overboard, or to burners, as the case may be. The fuel oil supply pump discharges to the suction of the fuel oil service pumps in each fire room. The service oil pumps discharge through an oil heater and meter to the boiler front. The fuel oil to the boiler fronts is arranged for recirculating in order that warm oil may be supplied the atomizers at starting. Thermometers and gauges are provided as well as a deck operated cutout valve for the fuel oil system from the service oil pumps and for steam to the service oil and supply oil pumps.

CARGO FUEL OIL PUMP.

Manufacturer	Worthington Pump & Mach. Corp.
Type	Vertical Duplex
Number per fire room	1
Cylinder, steam, diameter, inches	10
oil, diameter, inches	10¼
Stroke, inches	10
Nozzle, steam, diameter, inches	1½
exhaust, diameter, inches	2
suction, diameter, inches	8
discharge, diameter, inches	7

FUEL OIL SERVICE PUMP.

Manufacturer	Worthington Pump & Mach. Corp.
Type	Vertical Duplex
Number per fire room	2
Cylinder, steam, diameter, inches	5½
oil, diameter, inches	4
Stroke, inches	8
Nozzle, steam, diameter, inch	¾
exhaust, diameter, inch	1
suction, diameter, inches	2
discharge, diameter, inches	1½

FUEL OIL SUPPLY PUMP.

	Dobbin	*Whitney*
Manufacturer	Dow Pump & Diesel Engine Co.	Worthington Pump and Mach. Corp.

Type	Vertical simplex	Vertical simplex
Number, forward fire room	1	1
Cylinder, steam, diameter, inches	7½	6½
oil, diameter, inches	8½	7
Stroke, inches	8	8
Nozzle, steam, diameter, inches	1¼	¾
exhaust, diameter, inches	1½	1
suction, diameter, inches	3½	3½
discharge, diameter, inches	3	3

AUXILIARY FEED PUMP.

	Dobbin	*Whitney*
Manufacturer	Worthington Pump and Mach. Corp.	Dow Pump & Diesel Engine Co.
Type	Vertical duplex	Vertical simplex
Number per fire room	1	1
Cylinder, steam, diameter, inches	10	12
water, diameter, inches	7	7
Stroke, inches	10	12
Nozzle, steam, diameter, inches	2	2
exhaust, diameter, inches	2½	2½
suction, diameter, inches	4	4
discharge, diameter, inches	3	3

FIRE AND BILGE PUMP.

	Dobbin	*Whitney*
Manufacturer	Dow Pump & Diesel Engine Co.	Worthington Pump and Mach. Corp.
Type	Vertical simplex	Vertical simplex
Number per fire room	1	1
Number, in engine room	1	1
Cylinder, steam, diameter, inches	12	12
water, diameter, inches	14	10
Stroke	12	18
Nozzle, steam, diameter, inches	2	1½
exhaust, diameter, inches	2½	2
suction, diameter, inches	8	6
discharge, diameter, inches	7	5

MAIN FEED PUMP.

	Dobbin	*Whitney*
Manufacturer	Worthington Pump and Mach. Corp.	Worthington Pump and Mach. Corp.
Type	Vertical duplex	Vertical simplex
Number, engine room	2	2
Cylinder, steam, diameter, inches	10	15
water, diameter, inches	7	10
Stroke, inches	10	16
Nozzle, steam, diameter, inches	2	2
exhaust, diameter, inches	2½	2½
suction, diameter, inches	6	6
discharge, diameter, inches	4½	4½

FLUSHING PUMP.

	Dobbin	*Whitney*
Manufacturer	Worthington Pump and Mach. Corp.	Worthington Pump and Mach. Corp.
Type	Vertical simplex	Vertical simplex
Number, engine room	1	1
Cylinder, steam, diameter, inches	12	12
water, diameter, inches	10	10
Stroke, inches	18	18
Nozzle, steam, diameter, inches	1½	1½
exhaust, diameter, inches	2	2
suction, diameter, inches	6	6
discharge, diameter, inches	5	5

HEATING SYSTEM FOR CARGO AND FUEL OIL TANKS.

Steam coils are installed in the fuel oil service tanks in the inner bottom. Steam is supplied from the auxiliary steam system through reducing valves set at 50 pounds to distributing manifolds, one located in each fire room. There is also provided a relief valve and gauge at each manifold. Each fuel oil tank constitutes one circuit of the heating system. The drains from these circuits are led to steam traps located in the fire rooms. The traps discharge to an inspection tank, which

in turn drains to the main feed and filter tank. The heating surface is in the approximate proportion of 1½ square feet per ton of oil in the tank.

The outboard tanks, which are for the service and cargo oil, are provided with a coil heating system. Steam is supplied from the auxiliary steam system through a reducing valve set at 50 pounds. Relief valve and gauge is also installed. The lead of the supply pipe is run just below the main deck with extensions to manifolds, four on the port and four on the starboard side of the ship on the second deck. Valves on the manifold control the steam to the individual tanks. In addition to the control valve is a ¼-inch air vent valve in order to free the system of air. The drain from the coils is led to a common drain line, one in each pipe trunk, which terminates in a trap, one on each side of the after fire room. The traps discharge to an inspection tank, which in turn drains to the main feed and filter tank. The heating surface is in the approximate proportion of 0.6 square feet per ton of oil in the tank for the *Dobbin* and 1.5 square feet for the *Whitney*.

MAIN CONDENSING EQUIPMENT.

The main condenser is located on the port side of the engine room, and due to the limiting width in relation to the width of the turbines, was made with parallel vertical sides. The condenser is direct connected to the low pressure turbine by means of an exhaust trunk. The condenser drains through an air seal to the main air pump. A vacuum augmenter of the Parsons type is installed. A drain from the low pressure turbine is led to the vacuum augmenter and also to the air pump suction through a check valve in order to maintain adequate drainage for the low pressure turbines at all times. The main circulating pump is of the centrifugal type and draws from either a high or a low suction inlet and discharges overboard below the water line.

The tubes in the main condenser are spaced quite widely in the upper section, a little closer together in the second, and diagonally in the last section.

The installation is made in the *Dobbin* as described above and differs from the installation made in the *Whitney*, in that a centrifugal condensate pump is used for removing condensate and a radojet is installed for removing the non-condensible gases. Otherwise the installation corresponds to that made in the *Dobbin*.

The tubes are expanded at the end into which the water is directed and at the outlet end are packed with the usual gland and packing.

CONDENSER.

Manufacturer	Navy Yard
Cooling surface, square feet	8263
Tubes, number	3599
outside diameter, inch	⅝
thickness, mils	65
length, feet and inches	14– 2⅝
Distance between tube sheets, feet and inches	14– 0
Size of exhaust inlet, feet and inches	3 × 12– 5
Air pump suction, inches	16
Circulating water inlet, inches	18
outlet, inches	18

MAIN AIR PUMP ("DOBBIN" ONLY).

Manufacturer	Worthington Pump & Mach. Corp.
Type	Vertical simplex beam air pump
Cylinder, steam, diameter, inches	14
air, diameter, inches	28
Stroke, inches	18
Nozzle, steam, diameter, inches	2
exhaust, diameter, inches	2½
suction, diameter, inches	12
discharge, diameter, inches	10

MAIN CIRCULATING PUMP.

Manufacturer	Navy Yard
Type	Centrifugal
Low suction inlet, diameter, inches	18
High suction inlet, diameter, inches	14
Discharge outlet, diameter, inches	18

Impeller, diameter, feet and inches	3–10
Cylinder, steam, diameter, inches	11
Stroke, inches	12
Nozzle, steam, diameter, inches	3
exhaust, diameter, inches	3½

MAIN CONDENSATE PUMP ("WHITNEY" ONLY).

Manufacturer	C. H. Wheeler Mfg. Co.—Pump Terry Steam Turbine Co.—Turbine
Type	Centrifugal
Suction, diameter, inches	8
Discharge, diameter, inches	4
Impeller, diameter, inches	8½
Nozzle, steam, diameter, inches	2
exhaust, diameter, inches	3

VACUUM AUGMENTER ("DOBBIN" ONLY).

Manufacturer	Navy Yard
Cooling surface, square feet	101
Tubes, number	155
outside diameter, inch	⅝
thickness, mils	65
length, feet and inches	4– 2⅛
Nozzle, vapor inlet, diameter, inches	8½
drain outlet, diameter, inches	9
circulating water inlet, diameter, inches	4
outlet, diameter, inches	4
Jet, diameter, inch	5/16

RADOJET AIR EDUCTOR ("WHITNEY" ONLY).

Manufacturer	C. H. Wheeler Mfg. Co.
Type	H-3
Nozzle, air suction, diameter, inches	4½
steam, diameter, inches	1½
exhaust, diameter, inches	4

DYNAMO CONDENSING EQUIPMENT.

Below the dynamo platform is installed the dynamo condenser, which is equipped with independent circulating pump and air pump. Connection is made to this condenser for auxiliary exhaust through a back pressure valve.

CONDENSER.

Manufacturer	Navy Yard
Cooling surface, square feet	1820
Tubes, number	1141
outside diameter, inch	⅝
thickness, mils	65
length, feet and inches	9–11⅛
Length between tube sheets, feet and inches	9– 9
Thickness of tube sheets, inches	1
Dynamo exhaust nozzle, diameter, inches	22
Auxiliary exhaust nozzle, diameter, inches	10
Air pump nozzle, diameter, inches	7
circulating water inlet, diameter, inches	9
circulating water outlet, diameter, inches	9

AIR PUMP.

Manufacturer	Worthington Pump & Mach. Corp.
Cylinder, steam, diameter, inches	8
air, diameter, inches	16
Stroke, inches	12
Type	Vertical simplex
Nozzle, steam, inch, tap	1
exhaust, inches, tap	1¼
suction, diameter, inches	7
discharge, diameter, inches	6

CIRCULATING PUMP.

Manufacturer	Navy Yard
Suction, diameter, inches	9
Discharge, diameter, inches	9
Impeller, diameter, inches	21
Cylinder, steam, diameter, inches	6
stroke, inches	6
Nozzle, steam, diameter, inches	1½
exhaust, diameter, inches	2

AUXILIARY CONDENSER.

On the *Dobbin* there is installed an auxiliary condenser in the aft starboard side of the engine room. No similar condenser was installed on the *Whitney.* The condenser was manufactured by the Wheeler Condenser & Engineering Co., and is mounted over a combined air and circulating pump. The exhaust inlet is 14 inches in diameter.

MAIN FEED AND FILTER TANK.

The main feed and filter tank is located in the forward end of the engine room and is of the usual Department design, the upper part being arranged for filtering, the water passing through three filters arranged in series. In the bottom of the tank is admitted the high pressure trap drain which discharges into the first filter compartment of the feed and filter tank. Other trap drains discharge into this compartment through mufflers in order to eliminate the water hammer due to condensation of steam. The usual zincs, gauge glass, drain connections, and so on are installed. The capacity of the filter compartment is 390 gallons and the tank 1387 gallons for the *Dobbin* and 405 gallons and 1372 gallons respectively for the *Whitney.*

VAPOR CONDENSER.

There is placed above the main feed and filter tank a condenser in order to return the vapor released from the feed tank to the feed tank in condensed form. The condenser used was a distiller salvaged from the U. S. S. *Michigan,* when this vessel was scrapped in accordance with the Washington Limitation of Arms conference.

Circulating water is supplied as a cooling agent. In order to assist the operation of this system, if necessary, an exhaust fan is provided to draw the vapor into the distiller. The condensate from this distiller is led to the feed tank through a loop to provide a water seal. Extensions are led from this system to the glands of the main turbines.

SHAFTING.

The main shafting for the *Dobbin* and *Whitney* was manufactured by the Philadelphia Yard. The forgings for the *Whitney* were delivered to the Boston Yard rough machined. The material showed a tensile strength somewhat in excess of 60,000 pounds and an elongation somewhat in excess of 30 per

cent. One section of shaft is bored with an axial hole 4 inches in diameter for torsion dynamometer. This section is installed immediately aft of the reduction gear. The other sections are solid shafts. Shafting is arranged for drawing forward into the engine room, portable plates in the bulkheads being provided for this purpose. Stuffing boxes are installed where the shaft pierces watertight bulkheads. Brass sleeves are installed on the propeller shaft for bearings and to protect the shaft from corrosion. The bearings are of the usual lignum vitae type with stuffing box on the inner end of the stern tube bearing.

Sections, number	4
Flange, diameters, feet and inches	2– 6½
thickness, inches	3½
Journal diameters, inches	16¾
Shaft diameter, inches	16⅝
Taper coupling bolts, number	8
mean diameter, inches	3¾

TORSION DYNAMOMETER AND DETERMINATION OF SHEARING MODULUS OF ELASTICITY.

A Gary-Cummings torsion dynamometer is installed in the section of shaft directly aft of the reduction gear. The indicating end of the meter is located in the engine room. Shortly before the calibration of the meter for the *Dobbin* was undertaken instructions were received from the Bureau of Engineering that the data was to be used for computing the meter constant for the *Whitney* and also for the *Medusa* and *Holland*. The dimensions of the shafts of the *Dobbin* and *Whitney* are identical, and the shafts of the *Medusa* and *Holland* are identical but differ from the *Dobbin* and *Whitney* in that the axial hole is 7½ inches instead of 4, and in a few minor respects.

The Bureau further directed that the shearing modulus of elasticity be determined over a cylindrical length of the shaft. In order to obtain the latter data, pointers were installed on the

shaft when it was fitted up for calibration of the meter. These pointers were placed in a vertical position with a graduated arc placed at their lower end, the radius of the pointers being the maximum obtainable. No. 1 pointer was placed near the flange which was secured; No. 2 pointer was secured aft of a journal; No. 3 pointer was secured just forward of the next journal; and No. 4 pointer was installed at the free end of the shaft.

Weight arms were secured to the free end of the shaft so arranged that weights could be applied to twist the shaft. Observations of the meter reading were taken as the load was applied and as it was removed, and during this time the pointer readings were observed and recorded. As an additional check on the pointers, spirit levels were secured to the upper end of the pointers and the angles of twist read by the position of the bubble, the instruments being those regularly used to determine variation of gun roller paths. The data observed provided means for computing the torque in foot pounds on the shaft for one inch meter reading and the meter constant was computed by the following formula:

Equation 1

$$\text{Const.} = 2\pi/33000 \times \text{torque in foot pounds per inch meter reading.}$$

The shearing modulus of elasticity was determined by the use of the following formula:

Equation 2

$$G = 32\ Pal/\theta\pi\ (d_1^4 - d_2^4)$$

This equation was modified in order to use angles of twist in minutes and reduced to the following form:

Equation 3

$$G = 35016\ Pal/m\ (d_1^4 - d_2^4)$$

in which

G = shearing modulus of elasticity
P = Load applied in pounds
a = Radius at which P is applied in inches

l = Length of shaft under consideration in inches

θ = Angle through which the free end of the shaft was twisted (measured in arc of radius = 1)

m = Angle through which the free end of the shaft was twisted in minutes

d_1 = The outside diameter of the shaft in inches

d_2 = The inside diameter of the shaft in inches

The manufacturers state the meter reading in inches for one degree twist of shaft and a plot can be made showing the relation between these two quantities, the locus of the points being a straight line through the origin and the point plotted. This reading can be checked when the radii of the several actuating levers are known. With this information for any meter reading the corresponding angle of twist can be determined from the plot. In the same manner the relation between meter reading and torque in foot pounds can be plotted when the torque in foot pounds per inch meter reading is known. This curve also is a straight line through the origin and the point plotted. With these two curves on the same plot the relation between the three quantities can be readily determined. By means of the foregoing formula the following shearing moduli were computed:

Between pointers 1 and 4	11,764,000
Between pointers 2 and 3, cylindrical length of shaft	11,692,000
Between levels 1 and 3	11,692,000
By dynamometer	11,470,000
Mean	11,655,000

The results obtained by the meter were considered to be, after careful investigation, more reliable than the others in computing the meter constant for the *Medusa* and *Holland*. The experimental data was applied to the *Medusa* and *Holland* shafting by preparing a plot showing the relation between the meter reading and angular twist of the shaft as previously described.

Equation 3 was solved for P*a*, *m* being obtained from the plot for one inch meter reading. The result gave the torque in inch pounds for one inch meter reading which was readily converted to foot pounds. With this data a curve can be plotted showing the relation between meter readings and torque in foot pounds as previously described. The constant was computed by Equation 1. A check was made of these results by the method set forth in Sterling's Marine Engineers' Hand Book and the following results were obtained:

	Experimental	*Sterling*
Dobbin and *Whitney*	28.33	28.87
Medusa and *Holland*	24.66	25.12

STEADY BEARING.

The steady bearings for the line shafting are of the usual type, the housings being cast iron and the lower half fitted with a removable babbit lined brass bearing. The upper cap at the ends is also babbitted and fitted neatly on the shaft. Above is a reservoir of oil adapted for wick feed. Forward and aft of the bearing are drip reservoirs to catch lubricant from the bearing.

Bearings, number	6
Journal diameter, inches	16¾
Length, feet and inches	2– 3

PROPELLER.

The propeller was designed by the Bureau of Engineering and is of the four-bladed built up type, the material used throughout for the castings being manganese bronze. The blades fit into taper recesses in the hub and are held in position by forged steel bolts. The holes through the blades for securing bolts are oblong permitting adjustment of the blades for increasing or decreasing the pitch.

The working surface of the blades are machined to a true helicoidal surface. After the blades had been machined they were assembled on the hub of the propeller and balanced static-

ally, such excess material as was necessary to obtain static balance being removed from the backs of the blades. Following this operation the surfaces of the blades were ground and buffed smooth. The taper of the shaft through the hub is 1 inch per foot. A cap at the after end of the propeller covers the propeller shaft nut.

The Philadelphia Navy Yard made all the castings and machined all the blades for the *Dobbin* and *Whitney*. The hub for the *Whitney* was machined by the Boston Yard and the hub for the *Dobbin* by the Philadelphia Yard.

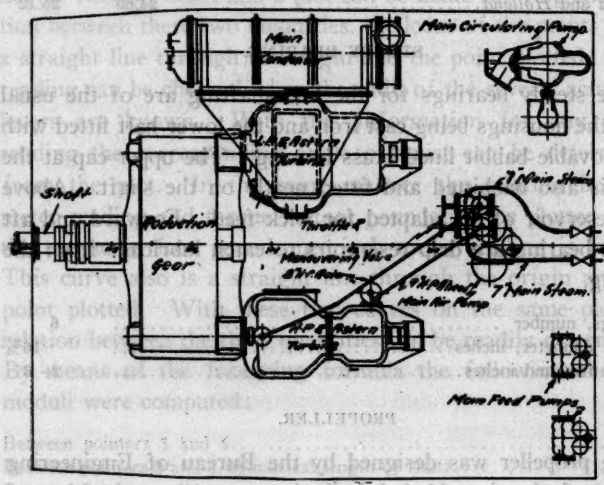


FIG. 4.

Propeller, diameter, feet and inches	16– 6
Direction of rotation looking forward	Clockwise
Projected area, square feet, four blades	90.12
Pitch, feet and inches	18– 4 7/32
PA ÷ DA	.405
Shaft horsepower	7000
Effective horsepower	4335
Revolutions	105

MAIN PROPELLING MACHINERY.

The main propelling machinery consists of a high and a low pressure turbine connected to a single propeller through reduction gearing. The arrangement of the engine room is shown in Fig. 4. In the high pressure turbine casing is a go-ahead and a go-astern turbine, and the same arrangement applies to the low pressure turbine, the ahead turbines being of the standard Parsons reaction type, while the astern units are of the impulse wheel type. The interconnecting piping between the two turbines is so arranged that if one turbine were to break down at sea blank flanges could be fitted into the piping and the turbine isolated.

The reduction gear is of the single reduction type and directly aft of it is located the main thrust bearing. The preliminary designs and specifications were furnished under contract by The Parsons Marine Steam Turbine Co., Ltd., duly approved by the Bureau of Engineering, for the turbines and reduction gear. The drawings were developed in consultation with representatives of that company and included the thrust bearing and main condensing equipment.

The work of manufacturing the turbines and reduction gear was divided in accordance with a mutual agreement between the Boston and Philadelphia Yards. The castings for the high pressure turbines and reduction gears were made for both ships by the Boston Yard, and the castings for the low pressure turbines were made for both ships by the Philadelphia Yard. The Philadelphia Yard manufactured the reduction gear and pinions for both ships. The forgings for both ships were made by the Philadelphia Yard, those for the Boston Yard being rough machined previous to delivery.

Casings.

The casings for the turbines are made of cast iron, divided on their horizontal axes. The joint between the casings is made metal to metal. Some of the flange bolts are "fitted"

and others are not. In the case of the high pressure turbine it was divided into two sections by a plane at right angles to the axis of the turbine, the joint between these two sections being male and female, and at the time the joints were being faced a groove was cut in each. After the sections had been bolted together manganisite paste was forced into one end of the groove until it was discharged from the other, after which the openings were closed with threaded plugs. A partition in the high pressure turbine casing separates the exhaust from the go-ahead and the go-astern turbines. By this arrangement the go-astern turbine is operating in a vacuum at all times while the go-ahead turbine is in operation. The exhausts from the low pressure go-ahead and go-astern turbines, however, are into a common chamber which is connected to the condenser.

Large steam belts are provided in the turbines for the admission chambers, of ample capacity to take care of any entrained water which may be carried along with the steam. A steam belt is provided between the first and second expansions for the steam from the by-pass valve. Drains from the high pressure turbine belts are led through stop valves to the exhaust chamber of the low pressure turbine. The stop valves are fitted with extension spindles to facilitate operation from the floor plate. The belt of the admission compartment of the low pressure turbine is of ample proportions.

In the high pressure astern turbine the nozzles are placed in the upper casing only; in the low pressure turbine they are placed in the upper and lower casings. The drain from the high pressure go-astern turbine is led to the exhaust chamber of the low pressure turbine and is kept open continuously, although a stop valve is provided in the drain in case of emergency. All steam and exhaust and lubricating oil connections of the turbines were placed in the lower casings with the exception of steam to the go-astern high pressure turbine, steam to the go-astern low pressure turbine, one nozzle being in the upper and one being in the lower casing and the exhaust to the condenser of the low pressure turbine. This arrange-

ment was selected in order to reduce to a minimum the amount of work involved in lifting turbine casings.

The casings, after the horizontal joint was made, were assembled and bored to the dimensions specified, grooved for the blading and the grooves serrated. After the blading had been installed the ends of the blades were machined to the specified diameters. In order to guard against distortion of the casings they were first rough machined and then thoroughly annealed, after which the final machining was done. Various pads are cast on the turbines for securing the lagging, which is made in sections to facilitate removal; for gauge connections, some of which are blank, but available for any special tests which might be conducted; for safety valves and for lifting gear fixtures. On the high pressure turbine steam belt is a relief valve set for 200 pounds, one on the go-ahead exhaust belt set for 15 pounds, and one on the low pressure turbine steam belt set for 10 pounds.

The forward ends of the turbine casings seat on guides which permit of axial movement due to thermal expansion of the turbine casings, the bolt holes for securing being made oblong to facilitate this, guides, however, prevent any lateral movement of the turbines. The after end of each turbine was fitted with a spigot which centered in a recess in the forward side of the reduction gear case, and this facilitated the alignment of the turbines and reduction gear. The flanges of the two are bolted together. The low pressure turbine, however, has additional supports in the vicinity of the exhaust belt for which suitable foundations were provided.

The astern nozzles were made by bending sheets composed of electrolytic copper and three to four per cent manganese, to the desired form for the vanes. These vanes were placed in the mold and cast into the nozzle plate which is composed of eighty-eight per cent copper, ten per cent zinc and two per cent tin. The nozzle plates for the *Dobbin* and *Whitney* were manufactured at the Philadelphia Yard, except the machining of the nozzle plates which was done by each Yard.

The following inlets and outlets are on the turbines:

	H.P.	*L.P.*
Steam inlet, diameter, inches		
Go-ahead	9	20
Go-astern	8	2–12½
Exhaust outlet, diameter, inches		
Go-ahead	20	36 × 72
Go-astern	17½	36 × 72

Thrust Bearings.

Thrust bearings manufactured by the Kingsbury Machine Works are installed on the forward ends of each rotor. These bearings are duplicates and take up any end thrust in the turbine rotor and serve to maintain the rotor in its correct axial position for all operating conditions. On the low pressure turbine forward of the bearing is installed a micrometer in order that measurements may be made of the axial position of the rotor. In the high pressure turbine is installed a micrometer of the usual type for determining the axial position of the dummy ring, and of course, the axial position of the rotor. As an additional check on the axial movement of the rotors a finger fitting was secured in the lower casing of each turbine which terminated near the steam deflecting flange on the rotor shaft, the distance between this finger and the flange being measured and recorded.

Data was furnished the ship as to the maximum allowable wear of the thrust bearing before renewal of the thrust shoes of the liners which are located forward and aft of the thrust bearing would be necessary.

Lubricant for the thrust bearings is supplied from below through two openings and is discharged out of the top of the bearing.

Bearing Data.

Number of shoes per bearing	6
Area per shoe in square inches	14.91
Total area for forward thrust, square inches	89.45
after thrust, square inches	89.45

Rotor Bearings.

The bearings are of the babbitted brass shell type, bored eccentric with the outside diameter to facilitate removal. An easing gear is provided to raise the shaft sufficiently for this purpose. During the final boring operations the boring tool faced off a surface concentric with the bearing surface to facilitate machining after rebabbitting.

Lubricant is introduced into a recess cut on each side at the horizontal dividing line of these bearings for nearly their full length and drains to a compartment below the bearing and thence to the sump.

Bridge gauges for checking the journal positions are provided and a label plate attached to each gauge which carries the data of the original position of the shaft. Pins are provided to check the horizontal distance between the gauge points and also its vertical measuring point.

General Journal Data.

High pressure turbine, diameter, inches	7
length, inches	11½
Low pressure turbine, diameter, inches	8
length, inches	13½

Turbine Glands.

The glands are of the labyrinth type designed to facilitate ready removal. Provision for introducing steam to seal the glands is provided. Steam from the auxiliary exhaust system is used on all the glands. The inside pressure on all the glands corresponds to the main condenser pressure, equalizer pipes being installed for this purpose.

Dummy Pistons.

Dummy pistons are installed on both turbines at the forward end and are fitted with the usual labyrinth packing, that on the high pressure being of the contact type, steam

tightness depending on the closeness of the labyrinth on the rotor and stator. For this reason a micrometer gauge is installed. The compartment forward of the dummy pistons is connected by equalizer piping to the exhaust compartments of the turbines.

Blading.

The blading for the turbines is made of a composition, copper 72 per cent, zinc 28 per cent. Special tools, jigs and fixtures were designed and built to facilitate the manufacture of this blading. The reaction and impulse blading was made by a rolling process in accordance with Parsons standard blade sections. The impulse blading was machined from stock and was made by the Philadelphia Yard for the *Dobbin* and *Whitney*.

The outer end of the impulse blades is milled to receive a shroud, one-half being fitted on the forward side of the blades, and the other half on the after side, the shroud being secured by silver solder. The lengths of the shrouds are short; or in other words, there are several lengths per row of blading. The root end of each blade is serrated to fit in the grooves in the rotor or stator, and tapered so that the blades shall set in a radial position. No separate distance pieces are used in this blading. A recess is provided to facilitate the introduction of the blades to the grooves, and the blades were slipped around until located in their correct position, the opening through which the blades were introduced being closed by a special method.

The reaction blading is supported at the tips by a binding strip and the longer blades by two binding strips secured to the blades by silver solder. The ends of all reaction blading are thinned and after installation were machined to the designed diameters.

Blading Data.

H.P. Turbine Reaction Blading.

Expansion No.	*Blade Section*	*Number of rows*	*Effective length of blade, inches*	*Tip clearance, inch*
1	130B	8	0.845	0.030
2	130B	8	1.095	0.030
3	130B	8	1.402	0.035
4	130B	8	1.777	0.035
5	130B	4	1.210	0.040
6	130B	4	1.585	0.040
7	130B	4	2.080	0.045
8	130B	4	2.705	0.045

H. P. Impulse Blading.

	Casing		Rotor	
Row No.	*Blade Section*	*Effective length, inches*	*Blade Section*	*Effective length, inches*
1	H607	1⅜	H607	1
2	H610	2⅛	H607	1¾
3	——	——	H616	2½

L.P. Turbine Reaction Blading.

Expansion No.	*Blade Section*	*Number of rows*	*Effective length of blade, inches*	*Tip clearance, inch*
1	240B	4	3.695	0.055
2	240B	4	4.815	0.060
3	250B	4	6.185	0.065
4	240B	2	4.180	0.070
5	250B	2	5.425	0.075
6	250B	1	6.295	0.080
7	250B	1	7.410	0.090
8	260B	1	8.775	0.100
9	260B	1	10.390	0.110
10	260B	1	10.390	0.110
11	260B	1	10.390	0.110

L.P. Impulse Blading.

	Casing		Rotor	
Row No.	*Blade Section*	*Effective length, inches*	*Blade Section*	*Effective length, inches*
1	H614	4⅛	H614	2¾
2	H620	6⅞	H614	5 13/16
3	——	——	H620	7¼

Turbine Guides.

Four cylindrical posts are installed forward and aft and secured into the lower casing of each turbine to guide the upper casing when it is being lifted. Portable guides are also provided for securing to the lower casing in order to guide the rotor while it is being lifted. Near the tops of these guides can be secured supports which will take the weight of the rotor.

Turbine Rotors.

The forgings for the rotors of the *Dobbin* and *Whitney* were manufactured by the Philadelphia Yard, and those for the *Whitney* were rough machined before delivery to the Boston Yard. The material showed a tensile strength somewhat in excess of 60,000 pounds and the yield point in the vicinity of 30,000 pounds. The high pressure rotor is made of one forging with the exception of the dummy ring and the wheel for the impulse blading which are separate pieces. The rotor is grooved for blading and labyrinth packing, the grooves for the blading being serrated. A wedge shaped groove is machined in the forward end of the dummy ring and a monel metal ring, for making micrometer measurements, is secured in this groove, the ring being forced in, in order to prevent the admission of any moisture which might cause corrosion and distort the micrometer face of the monel ring.

The low pressure turbine rotor is made of a solid forging, but wheels of disc construction are added for the dummy packing, reaction, and impulse blading. These wheels are machined to reduce their weight to a minimum, and are keyed and shrunk on. The rotor is grooved as described for the high pressure rotor.

After the blading had been installed and completed the rotors were subjected to dynamic balancing tests and were corrected for all dynamic unbalance, with the result that the turbines operate very smoothly and practically free from vibration under all conditions.

Maneuvering Platform.

The maneuvering platform is just forward of the turbines. The maneuvering valve is located for ready access from this platform, the valve being of the piston balanced type, one valve for the go-ahead condition and one alongside for the go-astern condition. Alongside the maneuvering valve are the engine telegraph, the log desk, and the gauge board. On this board were placed all the gauges required for the operation of the main turbines, the revolution indicator and the clock. On the aft end of the platform was installed the sound proof booth from which were led voice tubes to the various stations. A telephone set was installed in this booth.

MAIN REDUCTION GEAR.

The reduction gear is located aft of the high and low pressure turbines. These turbines are bolted to the forward side of the reduction gear case. The pinions are connected to the turbine rotors by means of a flexible coupling of the claw type. The axes of the pinions and main gear are in the same plane and the dividing line between the top and bottom casing is on this plane. The babbitted brass bearing shells are designed for ready removal, easing gear being provided to raise the shaft sufficiently for this purpose. When boring these shells the boring tool cut a surface concentric with the surface of the bearing to facilitate machining of the bearings after rebabbitting. The main bearing caps were designed to facilitate ready removal. The forward end of the shaft is fitted with a gear for operating the engine revolution counter gear. A micrometer is secured at the forward end of the shaft bearing to measure the axial position of the shaft. Information was provided the ship as to when it would be necessary to renew the liners of the thrust bearing due to wearing of the thrust bearing shoes, in order to return the shaft to its originally designed axial position.

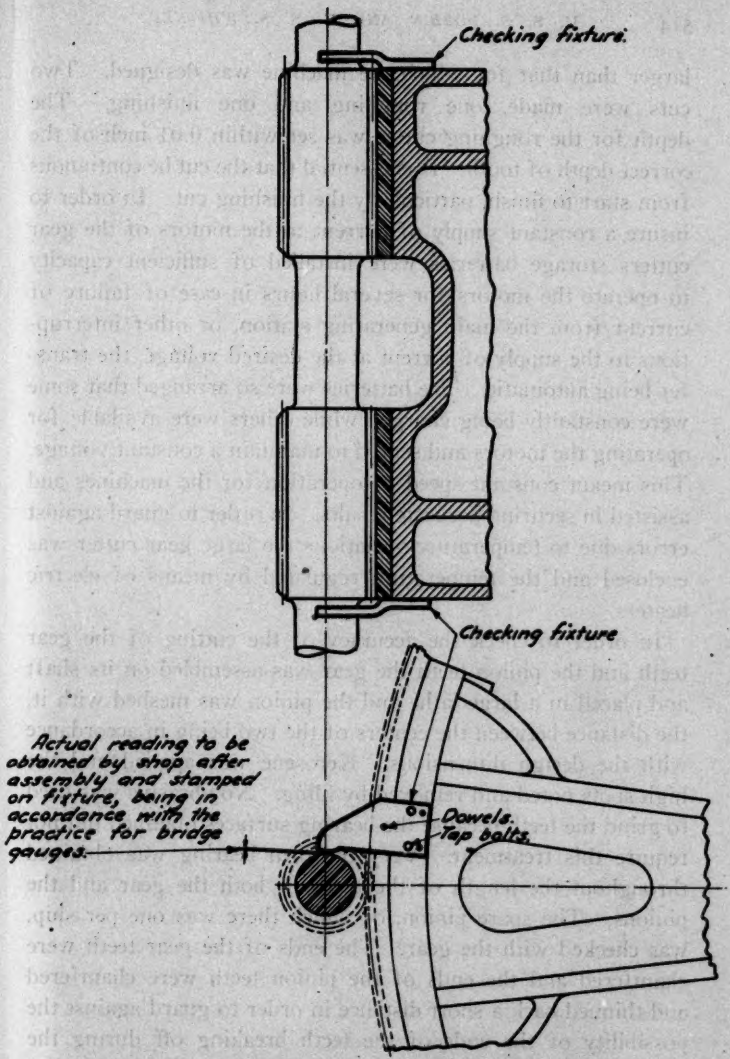
Lubricant is introduced into a recess cut on each side of the horizontal dividing line of the shaft bearings for nearly their full length. The oil drains to the gear case and from there to the sump tank in the inner bottom. Three sets of sprayers are provided for lubricating the teeth of the pinions and gear, two for the low pressure and one for the high pressure pinion. The sprayer for the high pressure pinion is located to discharge lubricant on to the teeth of the pinion as it goes into mesh with the gear for go-ahead condition. There are eight pipes per sprayer set. Three 1/8-inch holes are drilled in the end of each sprayer pipe. Each sprayer pipe can be removed for inspection independently.

Inspection plates are provided on the gear case in order to facilitate inspection of the equipment. A vent is installed at the top of the gear case and near it a warning plate to caution everyone to keep naked lights away.

The teeth of the large gear are cut in forged steel rims shrunk on a cast iron spider. Feet were cast on the spider and faced true with the axis of the gear after it had been assembled on its shaft to facilitate supporting and centering the gear blank on the gear cutter table.

The reduction gears and pinions of the *Dobbin* and *Whitney* are identical and both were manufactured by the Philadelphia Navy Yard. The Bureau of Engineering authorized the Philadelphia Yard to cut the reduction gears and pinions of the *Medusa* and *Holland,* which are identical, but of somewhat smaller dimensions than the corresponding details of the *Dobbin* and *Whitney.*

The cutting of the gears, which are the largest marine reduction gears cut in this country to date, presented many problems. The gear cutting machines were furnished under contract by the Parsons Marine Steam Turbine Co., Ltd. The parts of the machines were carefully checked previous to installation in the shops to insure accuracy of gear cutting and some modifications were made because the gear to be cut was somewhat


Checking fixture.
Checking fixture
Actual reading to be obtained by shop after assembly and stamped on fixture, being in accordance with the practice for bridge gauges.
Dowels.
Tap bolts.

U.S.S. DOBBIN – U.S.S. WHITNEY
MAIN REDUCTION GEAR.
METHOD OF CHECKING ALIGNMENT
OF PINIONS.

FIG. 5.

larger than that for which the machine was designed. Two cuts were made, one roughing and one finishing. The depth for the roughing cutter was set within 0.01 inch of the correct depth of tooth. It is essential that the cut be continuous from start to finish, particularly the finishing cut. In order to insure a constant supply of current to the motors of the gear cutters storage batteries were installed of sufficient capacity to operate the motors for several hours in case of failure of current from the main generating station, or other interruptions to the supply of current at the desired voltage, the transfer being automatic. The batteries were so arranged that some were constantly being charged while others were available for operating the motors and served to maintain a constant voltage. This meant constant speed of operation for the machines and assisted in securing accurate results. In order to guard against errors due to temperature variations the large gear cutter was enclosed and the temperature regulated by means of electric heaters.

In order to check the accuracy of the cutting of the gear teeth and the pinion teeth the gear was assembled on its shaft and placed in a large lathe and the pinion was meshed with it, the distance between the centers of the two being in accordance with the design dimensions. Kerosene was applied and the high spots noted and removed by filing. No abrasive was used to grind the teeth because the bearing surfaces obtained did not require this treatment. Very uniform bearing was obtained throughout the length of the teeth of both the gear and the pinions. The spare pinion, of which there was one per ship, was checked with the gear. The ends of the gear teeth were chamfered and the ends of the pinion teeth were chamfered and thinned back a short distance in order to guard against the possibility of the ends of the teeth breaking off during the operation of the gears and causing trouble. The gear after machining was mounted on its shaft and placed on parallel edges for balancing statically. The weights added were bolted in position and the ends of the bolts riveted over the nuts.

The time required to cut one face was about nine days, operating 24 hours per day. After one gear face had been cut the blank was turned over and the other face cut, the teeth being of the right and left hand spiral type. The successful completion of this work was largely due to the care and foresight of Master Machinist George J. Costello, who had charge of the work in the machine shop. A description of the machine shop problems involved prepared by the author was published in "The Iron Age," January 22, 1925, page 263.

It is essential that the gear case be accurately in line for successful operation of the gears and pinions. In order to accomplish this fixtures, extending over the pinion journals, were secured to the gear wheel. (See Fig. 5.) These fixtures were aligned by means of dowel pins and labeled for their proper positions. The distance between each fixture and journal was measured in the shop and recorded, the information being transmitted to those who had to do with the installation of the machinery. The casing was aligned in accordance with these fixtures in addition to alignment with the shafting before the foundation bolts were secured. These fixtures, together with the readings, were provided the ship, in order that the pinion alignment could be checked at any time. This method assured the installation of the pinions with their axes in the same plane as the axis of the reduction gear. The method was devised by Master Machinist Richard F. Cantwell.

The usual bridge gauges for checking the journals for vertical and horizontal position were provided and the readings recorded on the gauges. Pins were made to check the horizontal distance between the bridge gauge points and its vertical measuring point in order that gauge could be checked at any time for comparison with its original dimensions.

MAIN REDUCTION GEAR DATA.

Main shaft journals, diameter, inches	17
length (2 forward), inches	19½
length (1 aft), inches	12
Shaft horsepower transmitted	7000

	Gear	*Pinions*
Pitch circle, diameter, inches	146.183	9.903
Root diameter, inches	145.6054	9.3264
Outside diameter, inches	146.554	10.274
Width of each gear face, inches	21	21
Circular pitch, inch	0.676358	
Normal pitch, inch	0.583001	
Total depth of teeth, inch	0.4743	0.4738
Normal thickness of tooth on pitch circle, inch	0.2827	0.2815
Normal space between teeth, inch	0.3003	0.3015
Normal backlash, inch	0.0188	
Number of teeth	679	46
Angle at which teeth were cut	30 deg. 27 min. 41 sec.	
Revolutions per minute	105	1550
Weight of the gear wheel, pounds	60,000	
Weight of each pinion, pounds	1500	
Weight of gear shaft, pounds	13,000	
Tensile strength in pounds per square inch—		
Gear shaft and shrouds	60,000	
Pinion	95,000	
Yield point, pounds per square inch—		
Gear shaft and shrouds	30,000	
Pinion	65,000	

Main Thrust Bearing.

Aft of the reduction gear is located the main thrust bearing. The thrust bearing was manufactured by the Kingsbury Machine Works. The housing for the thrust bearing is mounted on a bridge casting which extended to the foundations of the reduction gear case and is bolted to flanges on the aft side of the reduction gear case. There is a center support for the bridge casting. The housing for the thrust bearing is secured to the bridge casting by means of fitted bolts and also to a flange on the after side of the reduction gear, the joint between these two being oil tight to provide for a drain of the oil from the after main bearing of the gear wheel and leakage from the thrust bearing compartment into the reduction gear case. Labyrinth packing is provided at each end of the thrust bearing compartment in order to prevent oil escaping along the shaft. The oil is admitted into this compartment through

two openings and discharged through the top. Aft of the after main bearing there is provided labyrinth packing in order to prevent the lubricating oil creeping along the shaft. The oil from this bearing is drained to the sump tank.

Main Thrust Bearing Data.

Bearing shoes, number	6
Area of shoe, square inches	55.8
Total area for forward thrust, square inches	334.8
astern thrust, square inches	334.8

TURNING GEAR, MAIN TURBINES.

A motor driven turning gear is provided for turning the turbines and reduction gear. The motor operates through two worm and wheel reductions. A sleeve coupling is used to connect the turning gear to the H. P. pinion. Provision is made for turning the turbines by hand. Gear guards are provided around the large worm wheel and the motor worm and worm wheel is enclosed in a case filled with grease. Ball bearing thrusts are installed throughout at each end of the worms.

Motor, H.P.	7½
Volts, D. C.	120
R. P. M.	500
Pitch circle, motor worm, diameter, inches	4
worm wheel, diameter, inches	11.9364
Pitch circle, main worm, diameter, inches	5
worm wheel, diameter, inches	29.2844

FORCED LUBRICATION SYSTEM MAIN TURBINES.

The main turbines and main reduction gear are fitted with a system of forced lubrication. On the after bulkhead of the engine room, port side, are located three lubricating oil pumps. Each pump is of ample capacity for ordinary cruising conditions. Oil is drawn from a sump tank located below the reduction gear and in the inner bottom. Independent suctions are led from the sump tank to each pump with an angle check valve installed on the top of the tank. The pumps discharge

to a manifold to which is secured a duplex oil strainer. The pumps are fitted with the usual relief valves, stop valves, gauges and air chambers. Below were placed oil drain trays to catch any oil which might drip from the pumps.

The discharge from the strainer is led to the settling tanks and to two oil coolers so piped that either or both or neither of the coolers can be used. Lubricating oil is led to the main thrust bearing, the sprayers and bearings of the reduction gear, and the bearings of the turbines. Oil from the turbines and thrust bearings is led independently to the sump tank. Oil from the reduction gear has an independent lead to the sump tank.

Gauges and thermometers are installed throughout the system both for supply and drain. Air vents are installed on top of all bearings which serve to indicate the supply of oil to these bearings. Oil to the rotor thrust bearings is supplied from below and discharged from the top of the bearing. A restriction plate is installed to limit the discharge from each thrust bearing. A similar arrangement is used for the main thrust bearing. Control valves of such design that they cannot be closed completely are placed in the supply to all the bearings.

Two pumps are installed on the after bulkhead on the starboard side to supply circulating water to the oil coolers. There is also an extension from the flushing system to the coolers. An extension from the forced lubrication system is run to the main circulating pump engine and an independent drain is run from the pump to the sump tank.

Means are provided for purifying the oil by a DeLaval separator, the oil to be separated being pumped into two settling tanks. The purified oil is discharged to the sump tank. Piping to the engine room lubricating oil tank provides means for filling the sump tank.

A pressure alarm system is installed consisting of a dead weight of such proportions that when the pressure in the system falls below 5 pounds electrical contact is made and a warn-

ing bell rung. The dead weight apparatus is installed near the rotor thrust bearings; in other words, at the end of the line.

The drains from the turbine bearings pass through an illuminated sight drain box. The sump tank, located in the inner bottom, has a capacity of about 1200 gallons. A float indicator is installed in this tank, indicator being located so that it is visible from the operating platform. A small hand pump is provided for draining the tank.

The oil inlets to the several bearings are as follows:

Turbine, main bearings, diameter, inch	¾
thrust bearings, diameter, inches	1½
Reduction gear sprayers (3), diameter, inches	2½
main bearings, diameter, inch	1
pinion bearings (3), diameter, inch	¾
main thrust bearing (2), diameter, inches	1½

LUBRICATING OIL COOLERS.

Manufacturer	Griscom-Russell Co.
Number	2
Square feet of cooling surface, each	300
Oil inlet, diameter, inches	4
outlet, diameter, inches	4
Circulating water inlet, diameter, inches	4
outlet, diameter, inches	4

LUBRICATING OIL PUMPS.

Manufacturer	Worthington Pump & Mach. Corp.
Number	3
Cylinder, steam, diameter, inches	8
oil, diameter, inches	10
Stroke, inches	12
Nozzle, steam, diameter, inch	1
exhaust, diameter, inches	1½
suction, diameter, inches	6
discharge, diameter, inches	5

CIRCULATING WATER PUMPS.

Manufacturer	Worthington Pump & Mach. Corp.
Number	2
Cylinder, steam, diameter, inches	8
water, diameter, inches	9

Stroke, inches	12
Nozzle, steam, diameter, inch	1
exhaust, diameter, inches	1½
suction, diameter, inches	5
discharge, diameter, inches	4

HANDLING GEAR FOR MAIN TURBINES AND REDUCTION GEAR.

In order to lift the different parts of the turbines and reduction gear for inspection and overhaul, a system of pad eyes is secured to the hull structure and differential hoists provided, the hoists being of the various weight capacities required for this service. The maximum lifting load is marked on each hoist. A nameplate is installed near each pad eye stating the maximum weight it will support. Pad eyes are installed over the turbo-generators for lifting weights.

SHOPS.

The shops in the *Dobbin* and *Whitney* differ in many respects, due to the fact that different machine tools with different operating characteristics were assigned the ships. Many of the tools have been furnished from excess stock on hand at various yards or naval bases and special tools which were not available from these sources were furnished on requisition. Furthermore, there have been differences in the designations of the shops, and since the ships have been in commission there have been many modifications made to meet the particular requirements of the ship.

Optical Shop.

This shop is located on the bridge deck, starboard side, between Frames 73 and 82. Adequate lighting is provided through ports on two sides of the shop. In the case of the *Dobbin* many changes were made by the ship's force, certain tools being removed and others installed. The equipment at present is in general as follows:

1—Precision lathe; 1—Watchmakers lathe; 1—Bench milling machine; 1—Bench grinder; 1—Bench drill; 1—Edge grinding machine; 1—Telescope collimator, Mark II; 1—Binocular collimator, Mark III, No. 11; 1—Watchmakers cabinet; 1—Electric oven; 1—Electric plating outfit; 1—Barnstead electric still.

Cabinets are provided for storing spare parts for repairs and for special tools. Benches adapted for the service were also installed, fitted with the usual facilities for this work.

On the *Whitney* the following tools were installed; otherwise, the shops are practically alike:

1—Milling machine; 1—Watchmakers cabinet; 1—Precision bench lathe; 1—Barnstead electric still; 1—Wet and dry grinder; 1—Tool room lathe; 1—Lathe; 1—Bench milling machine.

Foundry.

The foundry is located on the main deck on the port side between Frames 86 to 100½. There is a hatch at the forward port corner of the shop for access to the general machine shop located directly below. The foundries of the two ships are very similar, differing principally in minor changes made by the ship's force. The following equipment is installed:

1—Portable core oven; 1—1½-ton Cupola with blower; 2—Crucible oil burning furnaces for melting non-ferrous alloys.

Sand bins, coremakers benches and facilities for the storage of tools and flasks were provided. Overhead is a trolley system to facilitate the handling of heavy weights.

Coppersmith Shop.

This shop is located starboard of the foundry and there are a few minor differences between the shops of the two ships. The following equipment is installed:

Tinsmiths rolls; 35-ton Hydraulic pipe bender; Fuel oil burning furnace; Hand shear (*Whitney* only); Oxy-acetylene outfit; necessary workbenches and equipment usually furnished with benches.

On the *Dobbin* there was added a fuel oil burning forge and a bending slab. The fuel oil burning furnace for brazing was located below a skylight through which the end of the copper pipe could be extended if necessary. Overhead is a trolley system for the handling of weights.

Forge and Shipfitters' Shop.

This shop is located on the main deck between Frames 100½ and 114. The shops of the two ships are quite similar. The following equipment is installed:

1—100-pound Steam hammer; 2—Fuel oil buring furnaces; Water tank; 2—Anvils (*Dobbin*), 3 on the *Whitney*; 150-ton Hydraulic press.

On the *Dobbin* the bending slab installed in the shipfitters shop was transferred to the forge shop. A trolley is provided overhead for handling weights. A bench with the usual fixtures was furnished and provision was made for stowage of tools and material.

Starboard of the forge shop is the shipfitters shop. In this shop were installed a 24-inch radial drill press, grinder, and a small drill press, a hand punch, benches with the usual bench fixtures, facilities for the stowage of tools and material. Overhead is a trolley system for handling weights.

In the *Whitney* just aft of the shipfitters shop is a ship carpenter shop in which were installed a wood trimmer and benches for wood working.

Ship Carpenter Shop and Pattern Shop.

This shop on the *Whitney* is located as previously described. On the *Dobbin* it is located on the port side, main deck, between Frames 108 and 119 and was originally laid out as a

ship carpenter and pattern shop, but was changed by the ship to a ship carpenter shop.

A Universal woodworking machine was removed from the shop and installed in the starboard side forward of the machine shop, drill presses being removed to facilitate this arrangement. A work bench with the usual bench facilities was located outboard and near this machine. This outfit constituted the pattern shop. If pattern work beyond the facilities of this equipment is required the ship carpenter shop is available.

The following tools are installed in the ship carpenter shop:

Speed lathe; Combination rip saw; Band saw; Large lathe; Universal wood trimmer; 24-inch Planer and jointer.

Benches with vises and other tools were installed. Facilities for the stowage of tools and material were provided.

General Machine Shop and Electrical Shop.

The machine shops of the two ships differ on account of certain differences in equipment furnished as previously outlined. The shop is located on the second deck between Frames 57 to 90. On the starboard side between Frames 50 and 58 is located the shop office and drafting room. In a compartment in one corner of this office is located a blueprint machine and a blueprint washing tank and electrically heated dryer. In the room are cabinets for the storage of supplies.

On the port side between Frames 48 and 54 is the tool room in which are stored special tools and supplies. Forward and aft, port and starboard, are located the ventilating blowers which are fitted with thermo tanks. Overhead is a trolley system for transferring heavy weights. The trolley system extends through ports in the side of the ship for passing material into and out of the ship. Aft on the starboard side of the *Dobbin* is a motor boat engine shop, the machines originally installed in this space being located elsewhere.

Considerable difficulty was experienced in locating the machine tools due to the number and size of openings in the

second deck. These openings are for access to the torpedo work shop, the torpedo air compressors to facilitate overhaul of the cylinders, the cold storage rooms, the torpedo storage compartment, the ice machine room and the dry provision rooms. It was further necessary to maintain an open passageway forward and aft on the starboard side. Aft of the machine shop is the electrical repair shop on the port side between Frames 90 and 94, and aft of that is the gyro repair shop.

In these shops were located benches, some permanent and some portable, fitted with vises and the usual fixtures.

The equipment in the *Dobbin* shop is as follows:

General Machine Shop.

1—34-inch Vertical turret lathe; 1—2B Universal milling machine; 1—1½B Universal milling machine; 2—Precision lathes 8 inches × 22 inches; 1—No. 1 Milling machine; 2—Arbor presses; 1—Duplex milling machine; 1—36-inch × 60-inch × 16-foot Extension gap lathe; 1—Armature coil winder; 3—12-inch × 6-foot Engine lathes; 1—Armature winding machine; 1—20-inch × 10-foot Engine lathe; 1—Paper cutter; 1—Bake oven, small; 1—Bake oven, large; 1—Armature coil spacer; 1—Portable electric welder; 1—24-inch × 10-feet Engine lathe; 1—34-inch Vertical boring mill; 1—Horizontal boring, drilling and milling machine; 1—Centering machine; 1—20-inch × 12-foot Engine lathe; 1—14-inch × 6-foot Engine lathe; 1—Engraving machine; 3—16-inch × 8-foot Engine lathe; 1—2A Universal grinder; 2—12-inch × 2-inch Emery wheels; 1—Sellers tool grinding and shaping machine; 1—Twist drill grinder; 1—16-inch Sensitive drill; 1—Bench drill; 1—Cutter grinder; 1—28-inch Upright drill; 1—Straightening press; 1—16-inch Crank shaper; 1—Hack saw; 1—25-inch Crank shaper; 1—Double grinder; 1—Arbor press; 4—22½-inch Drill presses; 1—30-inch Radial drill; 1—2¼-inch × 24-inch Flat turret lathe; 1—24-inch × 36-inch × 8-foot Open side planer; 1—6-inch Metal saw; 1—Bench drill.

Electrical Repair Shop.

1—Coil winding machine; 1—Bench grinder; 1—Bench drill; 1—I. C. test panel; 1—General test panel; 1—Taping machine.

Gyro Shop.

Test panel and motor generator; Dynamotor for testing and repairing gyro.

The equipment of the *Whitney* shop is as follows:

General Machine Shop.

1—36-inch × 60-inch × 16-foot Extension gap lathe; 2—14-inch × 6-foot Engine lathes; 4—16-inch × 8-foot Engine lathes; 1—20-inch × 12-foot Engine lathe; 1—24-inch × 10-foot Engine lathe; 1—24-inch × 36-inch × 8-foot Open side planer; 1—34-inch Vertical turret lathe; 1—30-inch Vertical boring mill; 1—Horizontal boring, drilling and milling machine; 2—2A Universal milling machines; 1—24-inch Crank shaper; 1—16-inch Crank shaper; 1—28-inch Drill press; 2—23-inch Drill presses; 1—22-inch Drill press; 1—16-inch Sensitive drill; 1—30-inch Radial drill; 1—¼-inch to 3-inch Drill grinder; 1—¼-inch to 4-inch Centering machine; 1—6-inch Hack saw; 1—16-inch × 2-inch Wet tool grinder; 1—No. 2 Universal grinder; 1—6-inch Metal saw; 2—12-inch × 2-inch Emery grinders; 1—Buffing machine; 2—8-inch × 22-inch Precision lathes; 1—2¼-inch × 24-inch turret lathe; 2—Arbor presses; 1—Portable electric welder.

Electrical Shop.

1—18-inch × 10-foot Engine lathe; 1—Armature winding machine; 1—8-inch × 22-inch Precision lathe; 1—22½-inch Drill press; 1—Bench drill; 1—Bench grinder; 1—Coil winder; 1—Taping machine; 1—Paper cutter; 1—Coil

spreader; 1—Armature coil winder; 1—Bake oven, large; 1—Bake oven, small; 4—Armature stands; 1—Test panel, general; 1—Test panel, interior communication.

Gyro Repair Shop.

Test panels; Motor generator; Dynamotor.

TORPEDO WORK SHOP.

The torpedo work shop is located on the third deck between Frames 57 and 65. There are two independent plants, one port and one starboard, consisting of a torpedo tube, an air compressor and air storage reservoirs. The air piping is cross connected.

The air compressors are motor driven and are of the four-stage compression type. The compressors discharge through moisture separators to the air storage reservoirs. Air is discharged through separators from the air reservoirs, either direct or through reducing valves, as the design requires. A lead along the work bench provides outlets at 2500, 500 and 125 pounds pressure. On the discharge side of each reducing valve is a relief valve and gauge. Air is led to the torpedo handling room with outlets on the port and starboard sides for air at 2500 pounds pressure and on the port side for air at 125 pounds pressure. There is also a lead port and starboard to the ship's gas ejector system at 125 pounds pressure.

Since the ship was placed in commission the torpedo tubes which were installed for testing torpedoes have been stowed below and the space has been used for tools required for the work assigned this shop. Circulating water for the air compressors is drawn from a sea chest but an auxiliary supply may be obtained from the flushing system. A trolley system is provided for handling the torpedo tubes into and out of position and for handling torpedoes and parts of torpedoes.

The openings in the sides of the ship through which the torpedo tubes project when in position for firing are fitted

with watertight doors which are secured in position after the torpedo tubes had been withdrawn. The shop is fitted with the necessary facilities for the preservation and repair of torpedoes.

Compressor Data.

Compressor manufacturer	Ingersoll Rand Co.
Capacity in cubic feet per hour at 2500 pounds pressure	50
Cylinder, first stage, diameter, inches	13
second stage, diameter, inches	6½
third stage, diameter, inches	4¼
fourth stage, diameter, inches	1⅝
stroke, inches	8
Motor manufacturer	General Electric Co.
Motor H.P.	120
Volts, D. C.	120
R. P. M.	350 to 400
Starting panel	contactor type

"DOBBIN" OFFICIAL TRIALS.

The official trials of the *Dobbin* were conducted over the Rockland, Maine course and on the way from Rockland to Boston. No similar trials of the *Whitney* had been held at the time this article was prepared. The trials were conducted under the direction of the Board of Inspection and Survey after the ship had been placed in commission and was in operation by the ship's personnel. The standardization trials consisted of three runs at each speed selected, the data obtained being used to plot the relation between the speed of the ship in knots, revolutions of the propeller, and the corresponding shaft horsepower developed.

Run No.	*Speed*	*Average R.P.M.*	*Total S.H.P.*
1-N	6.99	47.55	552
2-S	7.46	47.75	599
3-N	7.56	47.71	554
Average Group No. 1	7.37	47.69	576
4-N	11.07	71.37	2184
5-S	10.94	71.31	2000
6-N	10.70	70.84	2113
Average Group No. 2	10.91	71.21	2074

Run No.	*Speed*	*Average R.P.M.*	*Total S.H.P.*
7–S	13.84	88.69	3852
8–N	12.88	86.93	3775
9–S	14.11	90.02	4047
Average Group No. 3	13.43	88.14	3862
10–N	14.63	99.59	5727
11–S	15.56	99.87	5594
12–N	14.17	99.70	5705
Average Group No. 4	14.98	99.76	5655
13–N	14.53	103.05	6326
14–S	16.75	104.72	6527
15–N	14.78	105.91	6871
Average Group No. 5	15.70	104.60	6563

Group average obtained by trapezoidal rule.

After the standardization trials had been completed six runs were made under the following conditions:

Full power for 4 hours;

14 knots for 4 hours;

14 knots for 2 hours with blower burners, the forced draft blowers being shut down;

12 knots for 2 hours with blower burners, the forced draft blowers being shut down;

Two runs at 10 knots for 4 hours each, one with one boiler, the other with two boilers, blower burners being used and forced draft blowers shut down in each case.

These tests provided some comparative data on fuel oil consumption with forced draft and with blower burners, although the runs were of rather short duration. One turbo-generator was in constant operation providing the necessary current for lighting, ventilation and the operation of the ship. In general the machinery performance was in accordance with the intent of the design and specifications. The following is the data of the trials:

Run	Full power	14 knots	14 knots blower burners	12 knots blower burners	10 knots blower burners 1 boiler	10 knots blower burners 2 boilers
Date	10/30/24	10/30/24	10/31/24	10/31/24	10/31/24	11/1/24
Duration, hours	4	4	2	2	4	4
R.P.M. of propeller	107.92	91.96	91.94	77.73	65.00	65.00
Knots, corresponding	16.19	13.95	13.95	11.92	9.99	9.99
Displacement, corresponding, tons	10,878	10,911	10,885	10,887	10,884	10,770
S.H.P., corresponding	7120*	4370*	4420*	2620*	1630*	1580†
No. boilers in use	2	2	2	2	1	2
No. burners in use per boiler	3	3	3	2 and 3	3	2 and 3
Vacuum, inches of Hg. mean	28.66	29.10	29.15	29.33	29.40	29.50
Slip of propeller, per cent, mean	17.17	16.22	16.24	15.33	15.14	15.14
Barometer inches of Hg. mean	30.40	30.38	30.41	30.41	30.33	...
Fuel oil, pounds, per hour	8620	6000	5720	4090	3330	3240
Fuel oil, pounds, per S.H.P. per hour	1.210	1.371	1.295	1.559	2.040	2.052
Pressure, pounds per square inch, mean:						
Boiler, F. R. No. 1	199	203	210	207	...	208
Boiler, F. R. No. 2	193	197	211	203	205	200
Turbine throttle, engine room	186	193	199	198	203	...
H.P. ahead steam chest	186	116	116	77	50	...
H.P. ahead second stage	153	96	94	56	43	...
L.P. ahead steam chest	21	15	17	10	7	...

*Calculated †From curve

Run	Full power	14 knots	14 knots blower burners	12 knots blower burners	10 knots blower burners 1 boiler	10 knots blower burners 2 boilers
Pressure, pounds per square inch, mean (Cont'd)						
Main feed pump discharge inboard	235	241	253	248	250	..
Main feed pump discharge outboard	235	241	253	256	307	...
Fuel oil at burners, F. R. No. 1	220	213	228	225	...	150
Fuel oil at burners, F. R. No. 2	221	245	190	154	180	140
Air, inches H_2O, F. R. No. 1	6.34	2.84	...	...	...	...
Air, inches H_2O, F. R. No 2	5.63	3.04	...	...	...	...
R.P.M. mean:						
Forced draft blower, F. R. No. 1, blower No. 1	1356	823	...	...	...	...
Forced draft blower, F. R. No. 1, blower No. 2	1341	794	...	...	...	...
Forced draft blower, F. R. No. 2, blower No. 3	1455	1125	...	...	...	...
Forced draft blower, F. R. No. 2, blower No. 4	1480	1125	...	...	...	...
Temperatures, mean:						
Lubricating oil to cooler	87	80	80	78	74	...
Lubricating oil from cooler	76	71	69	68	66	...
Lubricating oil from H. P. forward bearing	114	104	104	99	100	...

Run	Full power	14 knots	14 knots blower burners	12 knots blower burners	10 knots blower burners 1 boiler	10 knots blower burners 2 boilers
Temperatures, mean (Cont'd):						
Lubricating oil from H. P. aft bearing	117	115	115	106	103	...
Lubricating oil from L. P. forward bearing	114	119	110	108	106	...
Lubricating oil from L. P. aft bearing	121	111	120	117	117	...
Lubricating oil from H. P. rotor thrust	85	86	80	77	74	...
Lubricating oil from L. P. rotor thrust	86	82	80	76	75	...
Lubricating oil from main thrust	76	72	70	68	67	...
Injection water	50	49	50	50	50	58
Overboard discharge	77	67	71	67	62	76
Air pump discharge	70	69	54	53	54	...
Feed tank	84	82	84	79	90	62
Feed heater discharge	218	219	215	214	218	220
Fuel oil at burners, F. R. No. 1	132	121	128	123	...	122
Fuel oil at burners, F. R. No. 2	132	122	126	123	130	125
Engine room	80	80	82	80	84	...
Fire room No. 1	79	86	112	109	...	...
Fire room No. 2	78	82	92	92	95	...
Atmosphere	42	41	46	42	49	...

STEAM FLOW METERS ABOARD SHIP.

By Commander W. P. Beehler, U. S. N., Retired, Member.

Although steam flow meters have been widely used in power plants ashore for some years past, few installations have been made as yet aboard ship either in the Navy or Merchant Marine. The reason for this is difficult to understand when one appreciates the extraordinary benefits resulting from the use of such instruments and their indispensability to the economical and scientific management of power plants. In many quarters the impression seems to prevail that steam flow meters as now developed are difficult to install aboard ship to advantage, owing to the complicated nature of the steam leads in restricted spaces, and the alleged necessity for the use of a large number of instruments to obtain a complete record of the steam consumption of all the units of auxiliary machinery in operation. Further, it is argued, the motion of the ship in a seaway would seriously affect the accuracy of the readings. However, practical experience with such steam flow meters aboard the U. S. S. *Maryland* has shown that the above mentioned objections may be readily overcome, and that the error due to the motion of the ship is too slight to be apparent, besides which an error caused by a roll to starboard will be neutralized by the following roll to port.

The Bailey Fluid Meters, installed on board the U. S. S. *Maryland,* function on the pressure differential in the line resulting from a restricting orifice inserted at one of the flanged joints. The difference in pressure on the two sides of the orifice at specified points is made to operate on a mercury float, the movement of which is multiplied to actuate a moving arm with recording pen attached. A clockwork mechanism rotates

a chart on which the pen records the instantaneous rate of flow of the steam in pounds per hour; the float being so shaped that the chart graduations are equally spaced over the entire range thus facilitating the integration of the graph. Since the meter is constructed to give correct readings on the chart for a definite steam pressure and for steam of a particular "quality", any variations from the normal must be corrected by a factor obtained from the tables furnished with the meter. On vessels using superheated steam the normal degree of superheat will seldom vary sufficiently to require a correction factor to be applied and an occasional check will be necessary at infrequent intervals. A recording pressure gauge is believed to be essential aboard ship, and such gauge may be made to record directly on the steam flow chart or may be entirely separated from same. On board the *Maryland* the recording pressure gauge was an independent instrument located in the Engineers' Log Room. Since the installation was to some extent experimental it was deemed advisable to test the meter in its simplest form without the added complication of a pressure recording device. If accurate results are desired any variations in steam pressure in the line must be taken into consideration and the correction factor applied.

In making the installation of the steam flow meter on the *Maryland*, it was desired to locate the orifice plate at a point in the steam line where the piping was approximately straight for about 20 feet on each side of the flange. This was done to avoid the errors due to sharp bends in the pipe near the orifice. Such an ideal joint was found in the forward machinery space on the starboard side. The arrangement of the piping was such that by opening some valves on the line and closing others the steam to various combinations of auxiliary machinery could be metered. Since only one meter was available for the machinery spaces it was necessary to control the flow of steam through the auxiliary steam line in such a manner that only the steam to be measured was permitted to pass through the orifice. Fortunately, practically all of the

machinery in the forward space was duplicated in the after space so that it was nearly always possible to obtain a record of any unit desired by properly distributing the load between the two spaces. Had another meter been available for the after space the problem would certainly have been simplified.

The meter itself with recording apparatus was firmly secured to the thwartship flange on the bulkhead in a position to facilitate easy reading and access. Over 16 months' experience with these meters in various weather conditions in both the Pacific and Atlantic showed excellent results throughout. Even in severe weather the error due to the motion of the vessel was less than the thickness of the line made by the pen on the chart, or in other words, a small fraction of 1 per cent. The design of the instrument was sufficiently rugged to obviate mechanical difficulties and few adjustments were necessary during the entire period.

Two meters were installed on the *Maryland;* one in the machinery space forward to record the water rates of the various auxiliaries while the other was installed at boiler No. 7 and served to meter the total steam generated in this boiler together with a record of the airflow through the furnace. The boilers were designed for 35 degrees superheat which gave in practice about 30 degrees in the machinery space.

STEAM FLOW METER IN MACHINERY SPACE.

This meter was so located that steam could be metered to the turbo-generators (condensing and non-condensing sets), the steam driven ice machines, air compressors, main and auxiliary feed pumps, fuel oil pumps, fire and bilge pumps, main air pumps and air ejectors. In some cases one or more of these auxiliaries had to be metered at the same time, since at times it was not practicable to separate the steam flow to the various units. Under such circumstances the record of steam flow to the unit under consideration was obtained by subtracting the known steam consumption of the other units in operation from the total recorded on the chart. Practically

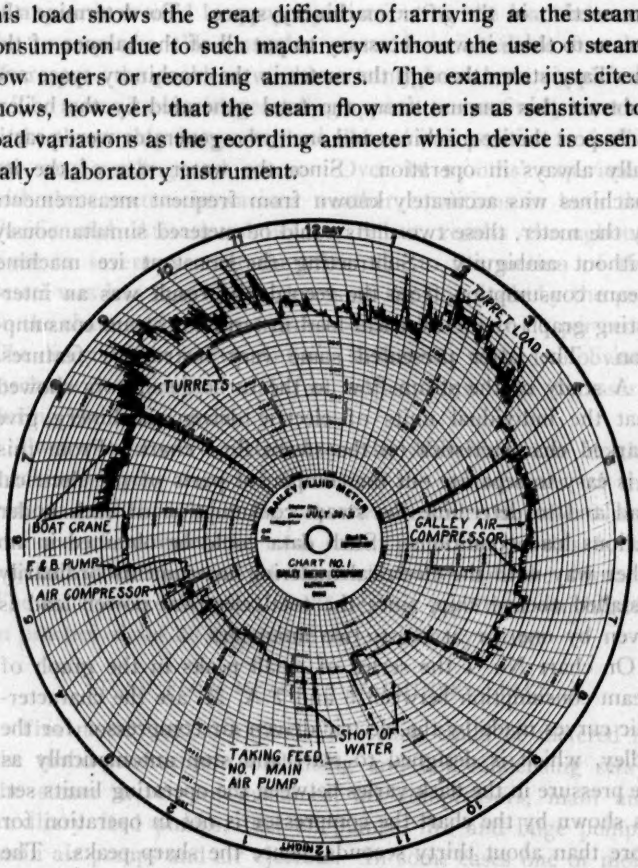
the only steam which could not be directly metered was that to the galley and pantries which was taken from the cross connection in the after machinery space. To determine the steam to these it was necessary to put all of the balance of the auxiliary steam through the meter in the machinery space and subtract this amount from the total generated by the boiler.

In port the ice machines and one turbo-generator were practically always in operation. Since the water rate of the ice machines was accurately known from frequent measurements by the meter, these two units could be metered simultaneously without ambiguity. Subtracting the constant ice machine steam consumption from the record, the result was an interesting graph of the electrical load in terms of steam consumption. The latter presented some very instructive features.

A study of the steam flow to the turbo-generators showed that the individual units of electric driven machinery give marked characteristics on the steam flow chart. From this it is easy to separate out the component parts of the electrical load and to determine the steam consumption of each under various load conditions. Such data could be obtained in no other way with the electrical metering instruments ordinarily installed aboard ship, since the total electrical power load is given by one, or at most, two ammeters.

On chart No. 1 the series of small peaks in the graph of steam consumption between 5 and 7 A. M. are the characteristic curves made by the electric driven air compressor for the galley, which is designed to start and stop automatically as the pressure in the flask varies between the operating limits set. As shown by the chart the compressor is not in operation for more than about thirty seconds, hence the sharp peaks. The sudden load variations caused by the operation of the electrical turret machinery are well illustrated by the same chart between the hours of 1:30 and 3:30 P. M. By subtracting the steam consumption due to the fairly constant lighting loads and blower loads during this period (obtained from the distribution room ammeters), the turret load in terms of steam per hour may be determined with fair accuracy.

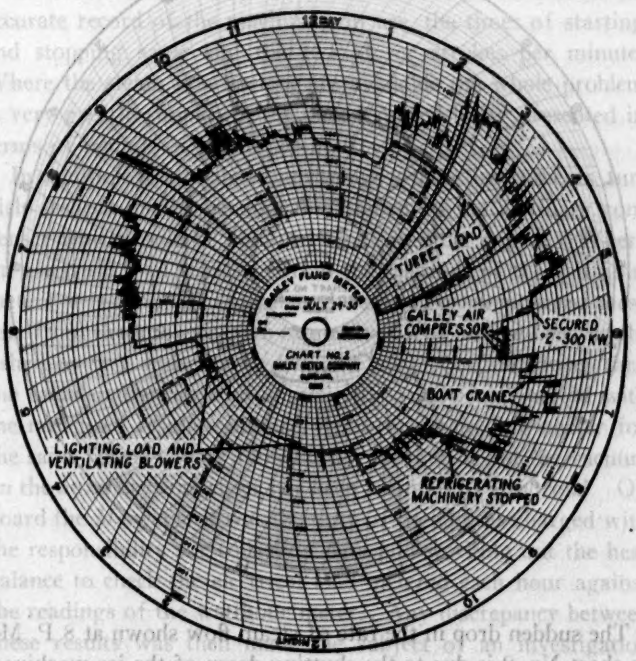
A typical boat crane load is shown on chart No. 1 between the hours of 6:15 and 7:15 A. M. The fluctuating nature of this load shows the great difficulty of arriving at the steam consumption due to such machinery without the use of steam flow meters or recording ammeters. The example just cited shows, however, that the steam flow meter is as sensitive to load variations as the recording ammeter which device is essentially a laboratory instrument.



On the same chart at 4 A. M. is indicated the typical curve of the large S. E. air compressors for charging the accumulators for the turrets, etc. This curve is distinguished from the curve due to reciprocating pumps in that the compressor curve has a sloping upper boundary due to gradual increase in

the steam consumption as the pressure in the flask builds up. This is followed by a sharp drop at the point where the power is cut off.

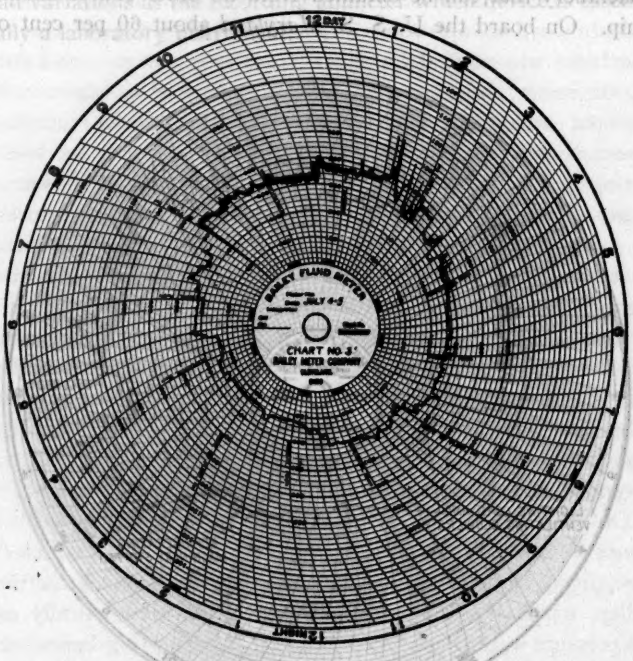
After a certain amount of experience with the steam flow meters it is found that the chart of steam consumption of the 300 K.W. sets gives a record of the operation of the electrical machinery that can be obtained from no other source aboard ship. On board the U. S. S. *Maryland* about 60 per cent of



the total load in port is electrical and the advantage to the Engineer Officer of such a complete and accurate record of the performance of the electrical plant is self-evident.

On chart No. 2 between 2 and 5 P. M. two condensing sets are being operated in parallel, one set being shut down at this

time, showing a decided drop in the rate of steam flow to the generators. Between midnight and 4 A. M. the load consists entirely of the lighting load and a fairly constant blower load. The small humps in the curve during this period are caused by the fire and bilge pumps which were in use for a few minutes during the watch.



The sudden drop in the rate of steam flow shown at 8 P. M. on chart No. 2 is due to the shutting down of the ice-machines which up to that time have been taking a supply of steam at the constant rate of 2400 pounds per hour. The valves were so arranged at this time that the total steam generated by the boiler was being passed through the meter. The large amount of steam used when the main air pumps are in operation is

well illustrated on chart No. 1 at the point indicated. In this case these pumps were in operation to supply extra feed to the boilers.

Chart No. 3 is a typical of charts obtained in port.

Every Engineer Officer, who has tried to work out a complete heat balance for the plant afloat, realizes the great difficulty of arriving at even an approximation of the steam consumption of auxiliaries which are in operation intermittently. With the best trained personnel it is difficult to obtain an accurate record of the machinery in use, the times of starting and stopping same and the R.P.M. or strokes per minute. Where the steam flow meters are available the whole problem is very greatly simplified, the data required being presented in terms of pounds of steam per hour.

In addition to the above advantages, these instruments furnish a check on the vacuum maintained on the auxiliary condenser which could be obtained from no other source. Where the electrical load is known from the readings of the watthour meters, it is possible to calculate the steam consumption of these sets from the known water rate curves which have been established by test and corrected by the meter.* Comparing then the calculated steam consumption for a particular period with the measured results obtained from the steam flow meter for the same period shows immediately whether or not the vacuum on the auxiliary condenser has been maintained at normal. On board the *Maryland* it was the duty of the Officer charged with the responsibility for collecting data and working out the heat balance to check the electrical load metered each hour against the readings of the watthour meter. Any discrepancy between these results was then made the subject of an investigation regarding the cause of the loss of vacuum. The necessity for such a check will be appreciated when it is realized that a loss of 5 inches of vacuum on the auxiliary condenser will result

* With a vacuum of 28.5 inches the water rate curves obtained from tests before installation of the 300 K.W. sets checked exactly with the results obtained by Steam flow meter.

in an increase in the water rate of the 300 K.W. sets of about 10 per cent. Since the drain from the evaporator coils was led to this condenser the reason for such vigilance is evident.

The effects of boiler priming are shown clearly on chart No. 1 at the points indicated by the arrows. Naturally such an occurrence cannot take place without the personnel on watch in the machinery space being aware of it, but the record on the steam flow chart is for the information of the Engineer Officer, who may then investigate the cause of such priming, whether the petty officer in charge of the watch has remembered to make a report of the circumstance or not.

With the record of the steam flow meter, boiler meter, watt-hour meter and the reports of machinery in operation, the Engineer Officer is in possession of sufficient data to work out a complete heat balance for the whole period or any part of same, without having to resort to estimates and laborious calculations unsupported by accurate data. Having reduced all factors to a common denomination of either B.T.U.'s or pounds of steam per hour, it becomes a relatively simple matter to determine the losses throughout the plant and to take the necessary measures to correct them. Without such complete and accurate data the Engineer Officer in charge must usually take a "shot in the dark" and hope for results. In other words, great efforts may be made to effect some slight economy in the plant while large avoidable losses may pass unnoticed due to the lack of accurate data on which to base a comprehensive analysis of the performance of all the units involved.

The Engineer Officer of the Watch, or the Officer having the days duty in port, was responsible for the proper regulation of the boilers to meet the load demands. To perform this duty effectively he must know what power will be required and the amount of steam necessary to meet these requirements. Failing this, a sudden load may be thrown on the boilers resulting in low steam pressure which may cause the air ejectors to break, followed by a loss of vacuum and a sudden further increase in the steam requirements. This vicious cycle once

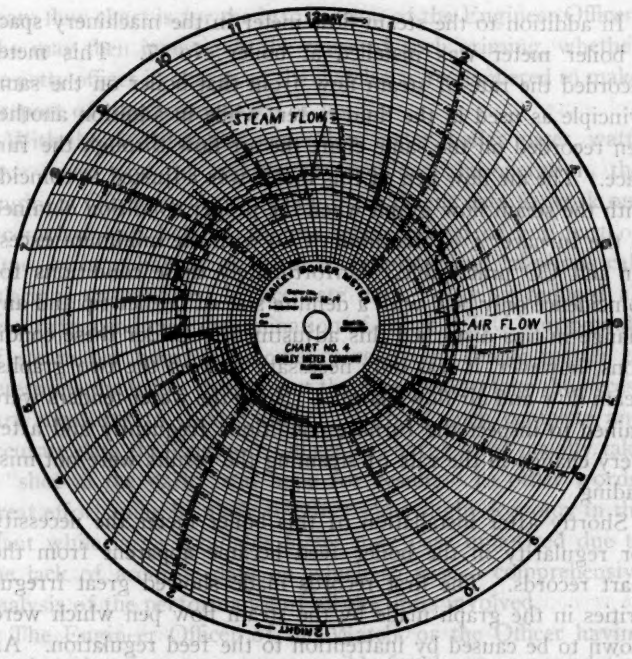
started means loss of all electrical power as well as a greatly increased fuel consumption. To help this situation the steam flow meters render valuable service in that the Officer of the Watch may keep himself accurately informed regarding the amount of steam required to meet the load conditions.

BOILER METER.

In addition to the steam flow meter in the machinery space a boiler meter was installed at boiler No. 7. This meter recorded the rate of steam flow from that boiler on the same principle as used in the fluid meter, while in addition another pen recorded on the same chart the air flow through the furnace. The air flow pen of this meter was adjusted to coincide with the steam flow graph when the proper condition obtained in the furnace; *i. e.*, maximum CO_2 with the minimum excess air and CO practicable. Unfortunately these conditions for coincidence only held with a definite size of burner tip and any change in tips disturbed this adjustment. Under these conditions, however, it was only necessary to keep the two graphs, steam flow and air flow, parallel. Since the water tenders were trained to inspect the furnace conditions frequently and after every change in the rate of steaming the results were not misleading.

Shortly after installation of this boiler meter the necessity for regularity in the boiler feed became apparent from the chart records. The first records made showed great irregularities in the graph made by the steam flow pen which were shown to be caused by inattention to the feed regulation. As soon as the attention of the water tenders was called to this fact an immediate improvement was noticed which continued until results were obtained with hand feed equalling the results obtained with the automatic feed regulators. In order that all water tenders might have the benefit of this training, they were put on watch in rotation on boiler No. 7 until they became proficient.

Naturally, every Engineer Officer appreciates the necessity for steady feed regulation, but the losses in boiler efficiency due to inattention to this feature can only be fully realized with such graphic records. It may be stated that the improvement in feed regulation resulted in an increase in boiler efficiency of about 5 to 6 per cent due to this factor alone. Further, the



attention of the water tenders being called to this fact by the meter the danger of priming due to high water in the boiler was reduced to a minimum.

Chart No. 4 shows the average performance in port; the various units of auxiliary machinery in operation producing the same characteristic curves as on the steam flow meter chart except to a smaller scale. The air flow line parallels the steam

flow line on this chart except at the points where the size of burner tips were changed. It will be noted that during the periods of steady load conditions the steam flow graph is steady showing good feed regulation. A characteristic boat crane load is very marked on this particular chart at about 7 A.M.

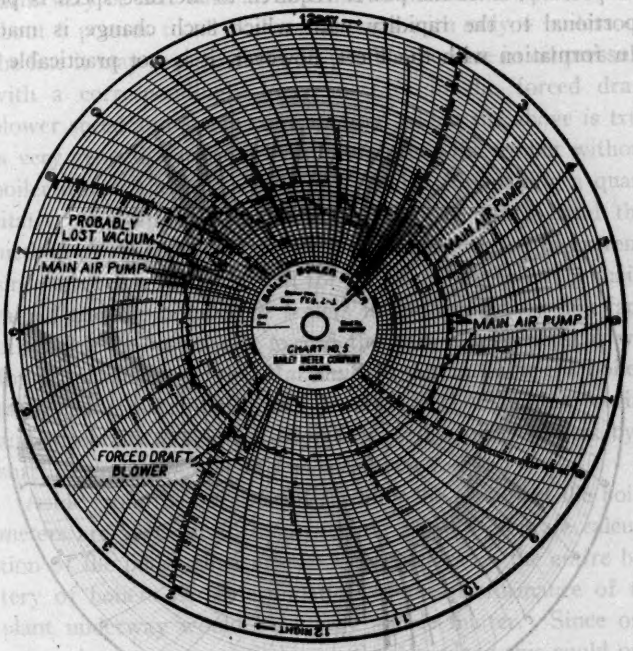
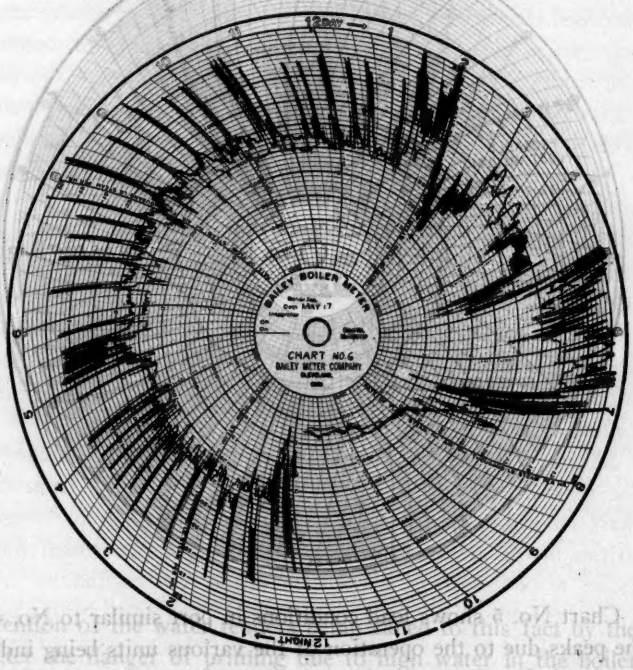


Chart No. 5 shows load conditions in port similar to No. 4, the peaks due to the operation of the various units being indicated. A few places will be noticed in this example where the air flow is excessive. This is due to speeding up the blowers for the purpose of cleaning the boiler tubes as a matter of routine before cutting out the boiler for a regular cleaning period.

The wide fluctuations in steam flow shown on chart No. 6 are due to the speed variations underway in formation with the Fleet. The sharp peaks indicate the large amount of power required to accelerate the ship when an increase of speed of over 5 R.P.M. is demanded. From the standpoint of economical operation, all changes in speed should be made as slowly as possible, since the power required to increase speed is proportional to the rapidity with which such change is made. In formation with the Fleet, however, it is not practicable to



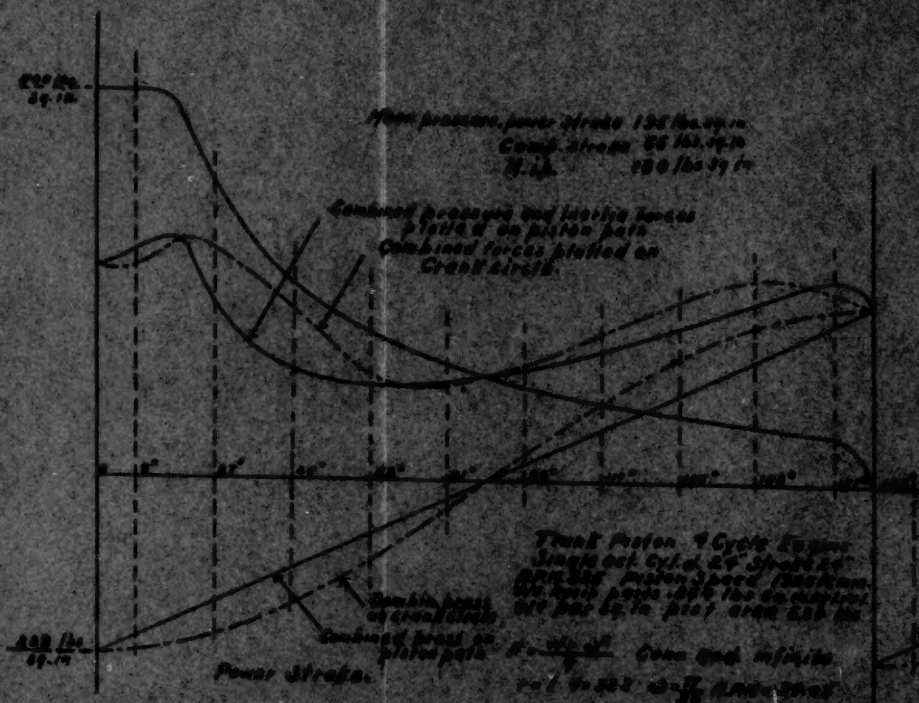
make these changes as slowly as would be required for the best boiler economy since in that case the ship would be out of position for long intervals of time. Signals from the bridge must be answered immediately and the steam controlled to meet the changes in power demanded.

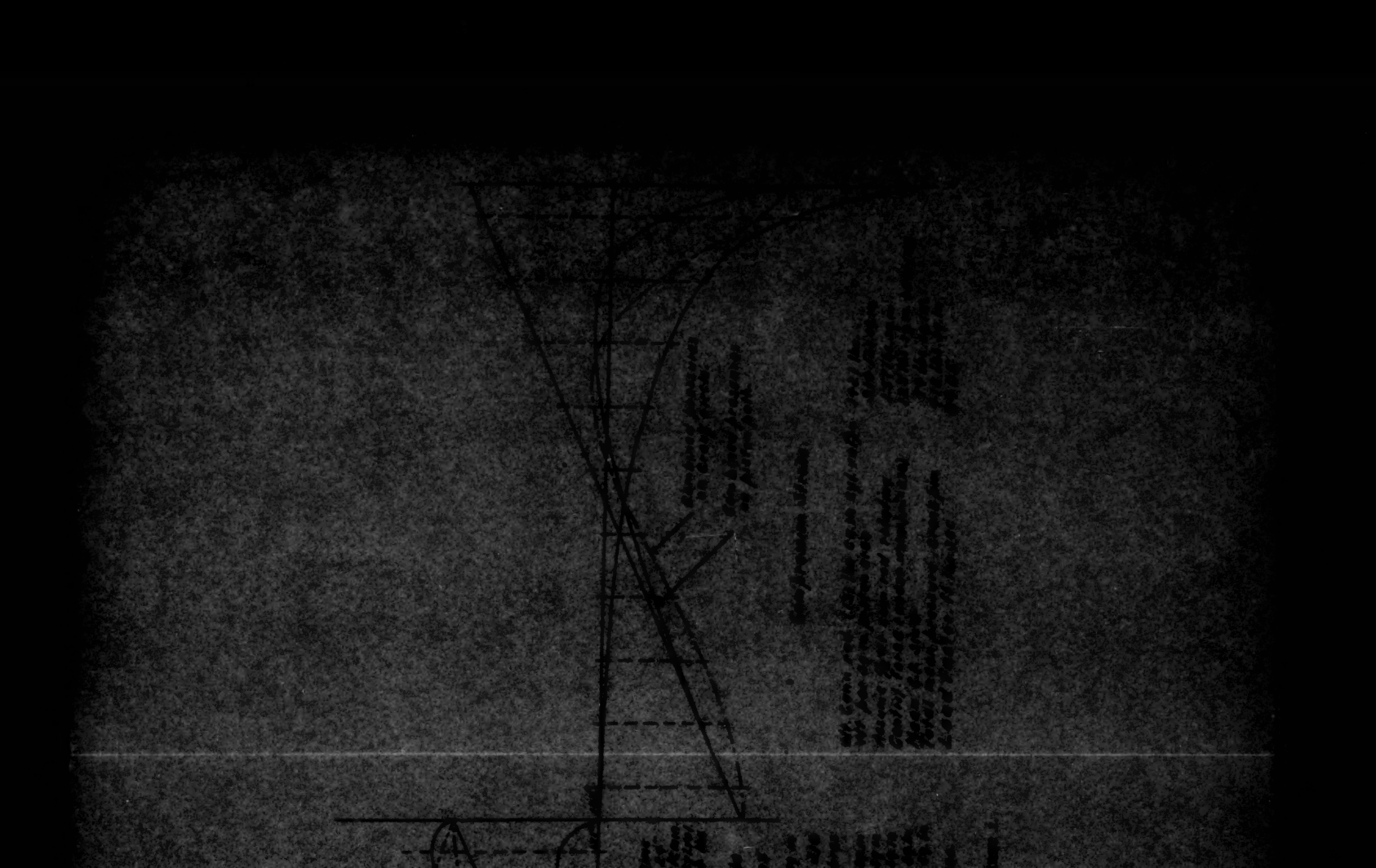
It may easily be shown that such sharp peaks in the steam flow line are accompanied by relatively low boiler efficiencies. The obvious method of reducing the amplitude of such peaks is to distribute the load over a greater number of boilers if practicable thus reducing the losses in the individual boilers. For example, with 7 boilers on the line a demand for 5 extra turns may be met with an increase in the fuel oil pressure of about 5 pounds to all boilers, whereas with only 4 boilers on the line it may require about 15 pounds increase in oil pressure with a correspondingly large increase in the forced draft blower speed and steam consumption. That the above is true is very generally realized by all Engineer Officers but without boiler meters there is no means available for making a quantitative analysis of this problem. On the other hand this method of reducing boiler losses cannot be carried to extreme since due regard must be had for the average speed to be maintained and the number of boilers in use limited by this fact; otherwise the losses will exceed the gain. In general it was found expedient on the *Maryland* to cut in sufficient boilers to meet the maximum power demands without undue forcing and to obtain economical operation at the lower powers by a shift to smaller burner tips as speed was reduced.

Aside from the question of training the personnel the boiler meters are very useful in permitting a rapid accurate calculation of the boiler efficiencies. With meters on the entire battery of boilers the determination of the performance of the plant underway would be a very simple matter. Since only one such meter was available on the *Maryland* this could only be determined indirectly and on the assumption that efficiencies of all the boilers on the line were approximately equal to the measured efficiency of boiler No. 7 which was equipped with a meter. (This assumption could only be made of course when the conditions at all the boilers on the line were about the same).

Probably the most vexing problem to the Engineer Officer aboard an Electric Drive Battleship is the question of the cost of fresh water in port in terms of fuel. At sea there is ample exhaust steam to supply the evaporators but the tanks available for the stowage of such "cheap" fresh water are limited and after a few days in port this reserve is usually exhausted. With low pressure evaporating plants there should be produced about 2 pounds of fresh water per pound of exhaust steam. This ratio is based however on the assumption of "dry" steam to the evaporator coils, which condition practically never obtains aboard ship particularly if reciprocating machinery is in operation. On the *Maryland* it was found necessary to install a home made calorimeter to determine the amount of moisture usually present in this exhaust steam. The result varied with the auxiliaries in use from about 8 to 13 per cent giving an average "quality" of about 89. The steam consumption of the various auxiliaries being known from the chart records a table was made out to show the amount of fresh water to be expected from the various combinations of auxiliaries for port service, due allowance being made for the quality of steam and the state of cleanliness of the evaporator coils. Thus the economy which would be expected from the installation of electric driven auxiliary machinery for port service could not be realized in general, since sufficient steam driven pumps, etc., had to be kept in operation to supply the required amount of exhaust steam to meet the demands for fresh water.

In general, both the steam flow meter and the boiler meter on the *Maryland* functioned well mechanically and required very little care and adjustment. As a means for training the personnel and checking the performance of duty of the water tenders and other Engineer personnel the meters were found to be extremely valuable. As a source of data for the solution of all problems pertaining to the operation and management of the plant they were indispensable. It is to be expected that in the near future such steam flow meters will be made a part of the regular equipment furnished Naval Vessels.


Mean pressure, power stroke 138 lbs. sq. in.
Comp. stroke 86 lbs. sq. in.
180 lbs sq. in.
Combined pressure and inertia forces plotted on piston path
Combined forces plotted on Crank circle
Power Stroke.



THE LIMIT OF PISTON SPEED IN A DOUBLE-ACTING TWO-CYCLE ENGINE AS AFFECTED BY BEARING PRESSURE.

By Captain A. M. Procter, U. S. N., Member.

In order to develop the high powers required in Naval engines under full power conditions, it is essential that such engines be run at the highest practicable piston speed.

This piston speed depends upon several factors. In this discussion only the effect of bearing pressure will be considered.

In order to establish a basis for comparison, combined pressure and inertia curves have been plotted, for the M. A. N. submarine engine; and the resultant load factor, for the crank pin bearing, has been computed. The results are shown on Sheet 1.

The load factor, which is the only true measure of bearing load, is the product of the mean bearing pressure and the rubbing velocity, expressed in foot-pounds per second.

For convenience in computing, a cylinder diameter of 24 inches, and a stroke of 24 inches, have been assumed. Since in similar engines, with the same weight of reciprocating parts per cubic inch cylinder volume, the inertia pressure per square inch piston area is the same for all cylinder diameters, the results are the same as if the correct cylinder diameter had been used.

As a check on the results obtained for the M. A. N. engine, an analysis of the performance of a Maybach aero engine, made under the direction of the British Air Ministry, is available.

These results are as follows:

Stroke, inches	7.09
Cylinder diameter, inches	6.50
B. H. P.	294
R. P. M.	1400
Piston speed, feet per minute	1654
Rubbing velocity, feet per second	15.85
Load factor	9270

This engine runs at a piston speed of 1400 feet per minute. Under the more normal conditions of 1300 feet piston speed, the M. A. N. engine is carrying fully as heavy a bearing load as the Maybach engine.

From a comparison of the above results with those of the M. A. N. engine, it is evident that the load factor of this engine, 8405 foot-pounds per second at normal full power, can safely be used as a basis for estimating the practicable piston speed for a very much larger engine.

Results for a two-cycle, double-acting, engine, with a cylinder of 40 inches, and a stroke of 48 inches, are shown on Sheet 2.

A value of 98 pounds per square inch has been assumed for the mean indicated pressure. This is within the limit of present practice for two-cycle engines.

The weight assumed for the reciprocating parts:—.42 pound per cubic inch cylinder volume and 20 pounds per square inch piston area—, is the result of an estimate based on an examination of the detailed weights of the following engines:

	Weight per cu. in. cyl. vol.
Maybach, aero, pound	.12
M. A. N., submarine, pound	.26
Maumee, 2-cycle, single-acting, pound	.62
Worthington, 4-cycle, single-acting, pound	.43
Worthington, 2-cycle, double-acting, pound	.81

9
0

5

inute.
speed,
g load

of the
f this
er, can
piston

linder
heet 2.
ed for
f pres-

pound
e inch
exami-

l. vol.
.12
.26
.62
.43
.81

	Load Factor for Inertia force	R.P.M	Piston Speed	Mean
Curve No. 1	136 lbs. sq. in.	81	648 ft. min	4
No. 2	213 lbs. sq. in.	102	816 ft. min.	5
No. 3	295 lbs. sq. in.	120	960 ft. min	5
No. 4	418 lbs. sq. in.	142	1130 ft. min.	6
No. 5	582 lbs. sq. in.	161	1288 ft. min.	7

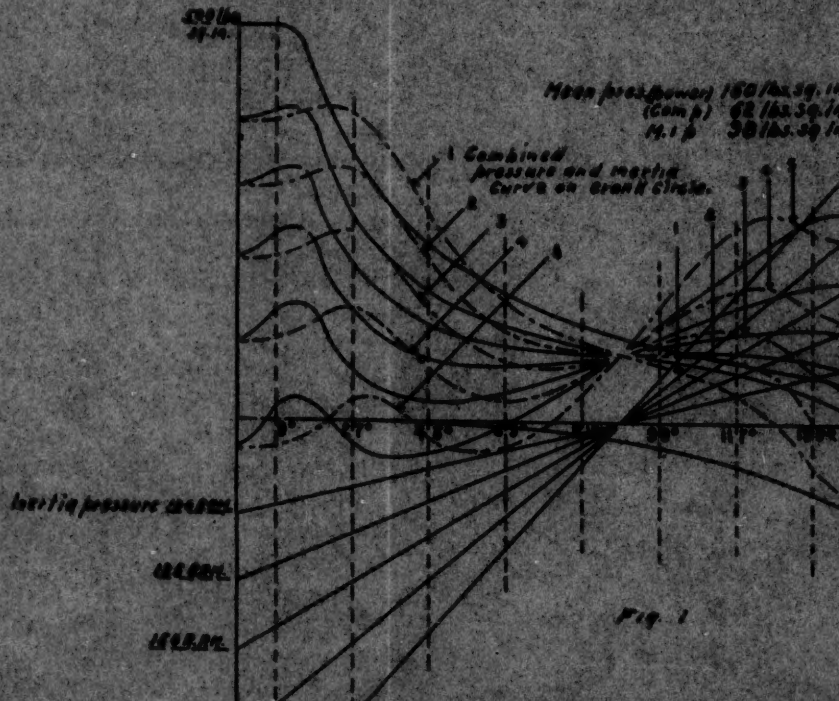


Fig. 1

2 Cycle Double Acting Engine

W. Reciprocating Parts 20 lbs.

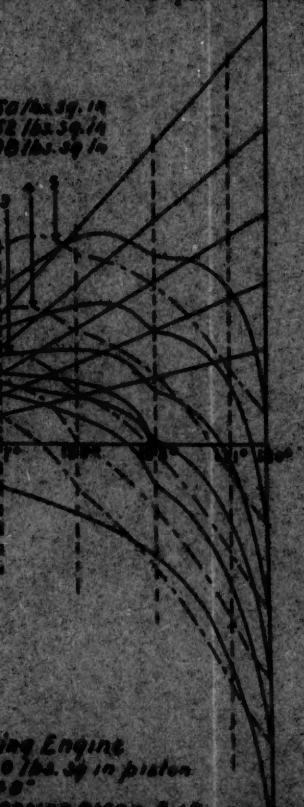
Cyl. diam. 10" Area 78"

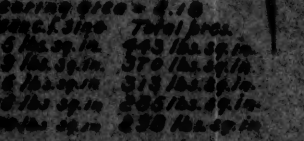
Ratio Piston area to Bearing area

R.P.M	Mean piston pres.	Mean bearing pres.	Mean C.I. pres.
100	132 lbs. sq. in.	418 lbs. sq. in.	26 lbs. sq. in.
125	182 lbs. sq. in.	231 lbs. sq. in.	33 lbs. sq. in.
150	118 lbs. sq. in.	467 lbs. sq. in.	66 lbs. sq. in.
175	96 lbs. sq. in.	2 03 lbs. sq. in.	70 lbs. sq. in.
200	91 lbs. sq. in.	190 lbs. sq. in.	100 lbs. sq. in.

R.P.M.	Total Bearing pres.	Rubbing Velocity	Load
100	145 lbs. sq. in.	10.49 ft. sec.	4647
125	370 lbs. sq. in.	13.06 ft. sec.	7038
150	313 lbs. sq. in.	16.70 ft. sec.	4916
175	286 lbs. sq. in.	18.34 ft. sec.	6287
200	338 lbs. sq. in.	20.91 ft. sec.	6251

Piston area	Mean Indicated pressure	C.F. Sine	Total bearing pressure	Rubbing Vel	Load factor
[illegible]	418 lbs. sq. in.	49 lbs. sq. in.	492 lbs. sq. in.	8.68 ft. sec.	37.92 ft. lbs. sec.
[illegible]	331 lbs. sq. in.	30 lbs. sq. in.	363 lbs. sq. in.	10.80 ft. sec.	3941 ft. lbs. sec.
[illegible]	237 lbs. sq. in.	28 lbs. sq. in.	310 lbs. sq. in.	12.56 ft. sec.	3894 ft. lbs. sec.
[illegible]	203 lbs. sq. in.	76 lbs. sq. in.	279 lbs. sq. in.	15.06 ft. sec.	4220 ft. lbs. sec.
[illegible]	138 lbs. sq. in.	100 lbs. sq. in.	238 lbs. sq. in.	16.80 ft. sec.	5015 ft. lbs. sec.





Load factor	Piston speed
1697 ft. lbs. sec.	800 ft. min.
1088 ft. lbs. sec.	1000 ft. min.
1814 ft. lbs. sec.	12 00 ft. min.
6227 ft. lbs. sec.	14 00 ft. min.
6281 ft. lbs. sec.	16 00 ft. min.

Diagram 2.

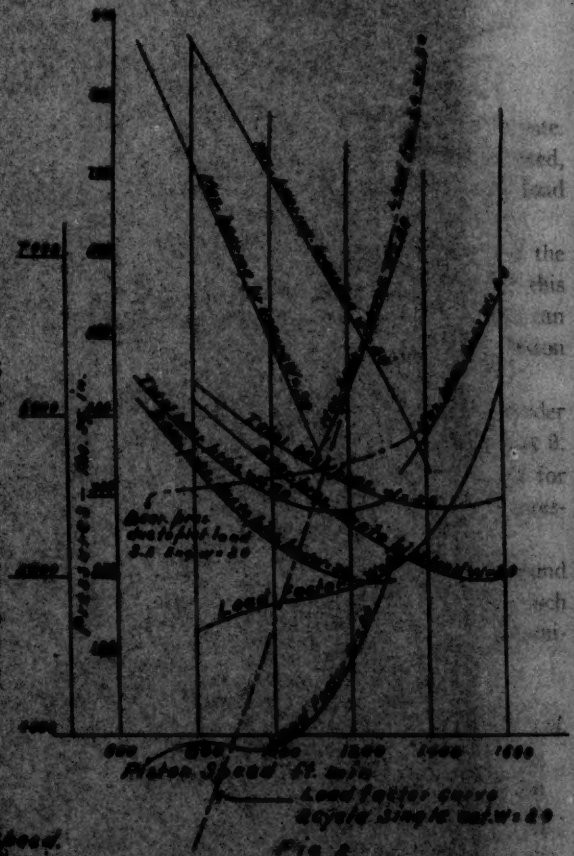


Fig 2.

In the above table, the weight for the Maybach engine has been multiplied by 1.33, the ratio between the maximum pressure of a Diesel engine and the maximum pressure of a constant volume explosion engine.

The last three engines are slow speed, heavy duty, engines; conservatively designed, without regard to weight.

The figure used in the computations on Sheet 2 is the minimum which might be expected in an engine where lightness of reciprocating parts is an important consideration.

For comparison with these results, bearing pressure and load factor curves have been computed for a double-acting, two-cycle engine with a weight of reciprocating parts equal to 30 pounds per square inch piston area, and for a double-acting, two-cycle engine with a weight of reciprocating parts equal to 20 pounds per square inch piston area.

These curves are all shown on Sheet 2, Fig. 1.

From an examination of these curves the following facts are noted:

1. For the double-acting engine, with a value of weight of reciprocating parts equal to 20 pounds, the curve of total bearing pressure shows a marked decrease until a piston speed of 1600 feet per minute is reached.

2. For the double-acting engine, with a value of weight of reciprocating parts equal to 30 pounds, the curve of total bearing pressure shows a materially smaller pressure than the above, until a piston speed of about 1400 feet per minute is reached.

3. For the double-acting engine, with a value of weight of reciprocating parts equal to 20 pounds, the value of the load factor at 1600 feet per minute is 6231 foot-pounds per second; which is but 74 per cent of that of the M. A. N. engine at normal full load.

The superiority of the two-cycle, double-acting engine, over all other types, is apparent.

It is probable that other factors, the most important of which is the necessity of maintaining a good propeller efficiency, will limit the speed to about 1300 feet piston speed.

At that speed, the load factor is about 5000 foot-pounds per second, which is 60 per cent of that of the M. A. N. engine.

At this piston speed, where the two load factor curves cross, the advantage to be derived will be in the shorter bearings required; resulting in a materially stiffer shaft, and in a material decrease in the bending moments on the cylinder housing, due to the reduced distance between cylinder centers.

LUBRICATING GREASE.

By Lieut. Commander G. B. Vroom, U. S. N., Member.

Correct lubrication is an operating problem of first importance, and one that gets less thought than its importance warrants. Scientifically, it is receiving more and more attention, in the effort to determine the exact actions and reactions that take place in the lubricant itself, under the effects of heat, speed and pressure; to the end that measurable physical and chemical characteristics of classes of lubricants may be intelligently selected to fulfill definite working conditions. Operating engineers, and the powers above them that foot the operating expenses and scan the plant overhead, are undoubtedly coming to realize the importance of the problem, as speeds, temperatures and weights of bearings keep pace with advances in design and metallurgy. Each branch of engineering has its peculiar conditions to meet, particularly in Diesel practice.

Automotive machinery, marine machinery units, and ball bearings and thrusts are examples of important applications of grease lubrication.

There is one class of lubricants whose technology is at a standstill, in comparison with oils; and that class is—Grease; a humble member of the engineering family. Yet thousands of tons of it are used; hardly a unit of the power plant can avoid the use of the grease cup, or its equivalent. There are many lubricating requirements that cannot employ fluids; and yet the necessity for correct and ample lubrication is absolute.

"Grease is grease" and nothing more to most people, so far as its suitability to meet specific conditions is concerned; but it should take its place, in technical importance, alongside the

oil whose substitute it necessarily is. In the last analysis, it is the oil content of the grease that performs the lubrication, and therefore it follows that the grease should be made of an oil suited to the conditions of temperature, speed, and pressure. Oils are manufactured and marketed to meet all recognized operating conditions; producers maintain extensive laboratory and field services to study the needs of consumers and to assist them in intelligent selection of oils to meet their needs. Greases, however, are still largely by-products, and rarely are made of oils higher in viscosity than spindle oils. They are graded simply on *consistency;* in other words, upon the percentage of soap content, which has nothing to do with the lubricating qualities of the grease after it is melted and flows to the bearing. Consistency and flow point are very important factors *before* the grease reaches the bearing, because in many types of feed they determine the amount of lubricant that is supplied, in proportion to the temperature of the grease, and type of container.

It is not simply a question of outwardly apparent increased efficiencies of individual units, but of prospective savings in replacement and repair costs, and avoidance of costly tie-ups.

Greases should be manufactured and selected for use, with regard to:

(*a*) Consistency.
(*b*) Flow point.
(*c*) Characteristics of contained oil.
(*d*) Freedom from impurities and fillers.

The characteristics of the contained oil should conform in all respects to those of an oil that would be specified for similar operating conditions where oil, instead of grease, could be used. These characteristics are of two classes: chemical and physical. It is important, for instance, to guard a bearing from a grease whose oil is poorly refined and high in acidity; it is equally important that the oil shall be of a viscosity to stand up under the working conditions to be met.

Consistency and flow point should be prescribed to meet the working temperature conditions, the rate of feed desired, and the design of the feed system.

That the grease should be free of impurities, fillers and injurious substances, goes without saying.

To raise the standard of practice in the use of greases, a definite series, following the scheme of lubricating oil classification, would have to be adopted. This is illustrated in the following series; but it is to be understood, of course, that the figures are illustrative only, and are not proposed as the suggested series:

GREASES—MINERAL.

Contained Oil.	*Flow Point of Grease.* degrees F.	*Consistency of Grease.*
Viscosity Saybolt at	140–150	Soft
130 degrees F.	165–175	Medium
100–115 seconds	210–220	Hard
Viscosity Saybolt at	140–150	Soft
130 degrees F.	165–175	Medium
125–145 seconds	210–220	Hard
Viscosity Saybolt at	140–150	Soft
130 degrees F.	165–175	Medium
180–200 seconds	210–220	Hard

A recent analysis of fifty samples of greases brought to light a considerable amount of comparative data; and accentuated the present want of standardization, not alone in attention to the viscosity of the constituent oil, but even a want of uniformity in grading, so that a customer who ordered a "No. 1" grease from one producer, might, in changing his source of supply, get a grease from the other end of the consistency scale, even though he ordered "No. 1."

Of the fifty samples, all but nine were made of an oil of viscosity 65—75 seconds Saybolt at 130 degrees F., or lower. Obviously, greases made of these low viscosity oils would give a very thin lubricating film.

The *flow point* of grease, according to this analysis, is no indication of its hardness, as many of the *soft* greases had as high flow points as the *hard* greases. There are no standard limits of hardness for different grades of greases. But, since both *flow point* and *consistency* are of actual value in determining the suitability of a grease for a given use, definite standards should be adopted, so that a grease classed as soft, medium or hard, will be within the standard limits for its grade, regardless of the source of supply.

The proposed classification gives nine grades, with respect to physical characteristics, from which to select the proper lubricant with due regard to *all* physical conditions, both in the bearing, and in the feed system. *In* the bearing, the *viscosity* of the contained oil would be the governing factor; in the feed system, *consistency* and *flow point*. In other words, get the proper amount of the correct oil to the bearing.

Chemically, the grease should be free from deleterious impurities. The contained oil should be well refined and contain no mineral acids, sulphur, or fatty oils; and should have specified flash, fire and pour points.

II.

The first step in the formulation of a scientific practice, is the determination, by experimental laboratory data, of the behavior of greases made up to conform to known operating conditions, in respect to bearing temperature and pressure, and the temperatures of cups, both gravity and continuous feed types. The temperature of the cup, as it varies above and below the melting point of the grease, controls to a large extent the rate of feed of grease to the bearing, and hence is an important factor in the lubrication of the bearing. The temperature of the cup and its contents depends upon two components —temperature of the surrounding air, and heat conducted from the bearing.

Figure I shows diagrammatically a grease testing machine. It consists essentially of a journal driven by a motor, so arranged that the bearing may be spring loaded, and any de-

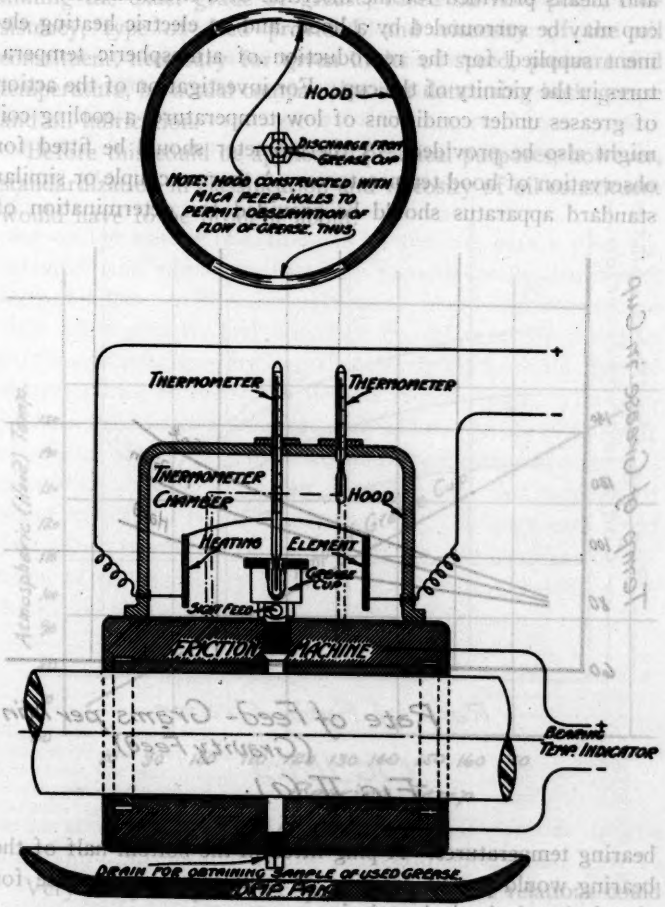

HOOD
DISCHARGE FROM GREASE CUP
NOTE: HOOD CONSTRUCTED WITH MICA PEEP-HOLES TO PERMIT OBSERVATION OF FLOW OF GREASE. THIS.
THERMOMETER
THERMOMETER
THERMOMETER CHAMBER
HOOD
HEATING ELEMENT
GREASE CUP
SCREW FEED
FRICTION MACHINE
BEARING TEMP. INDICATOR
DRAIN FOR OBTAINING SAMPLE OF USED GREASE.
DRIP PAN

FIG. 1.

sired pressure applied. A grease cup is mounted, as shown, and means provided for the insertion of a thermometer. The cup may be surrounded by a hood, and an electric heating element supplied for the reproduction of atmospheric temperatures in the vicinity of the cup. For investigation of the action of greases under conditions of low temperature, a cooling coil might also be provided. A thermometer should be fitted for observation of hood temperatures. A thermo couple or similar standard apparatus should be provided for determination of

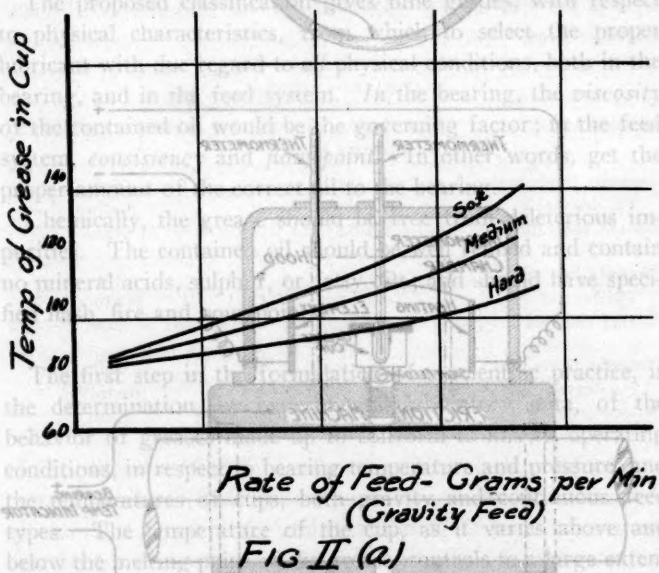


FIG II (a)

bearing temperatures. A plug fitted in the bottom half of the bearing would permit draining of samples of used grease for chemical and physical analysis.

Coefficients of friction should be calculated as in the case of tests of oils.

The possible lines of investigation, independent or related, with the apparatus of Figure I are largely apparent; it is sug-

gested, however, that its ultimate usefulness would be in determining the exact grade of grease, based on flow point, consistency, type of feed provided, and viscosity of the oil constituent, necessary for given limits of speed, pressure and temperature, by actual comparative test data using both grease and oil lubrication.

Before this could be applied to practical purposes, however, standardization in classification and viscosity of oil constituent would have to be brought about.

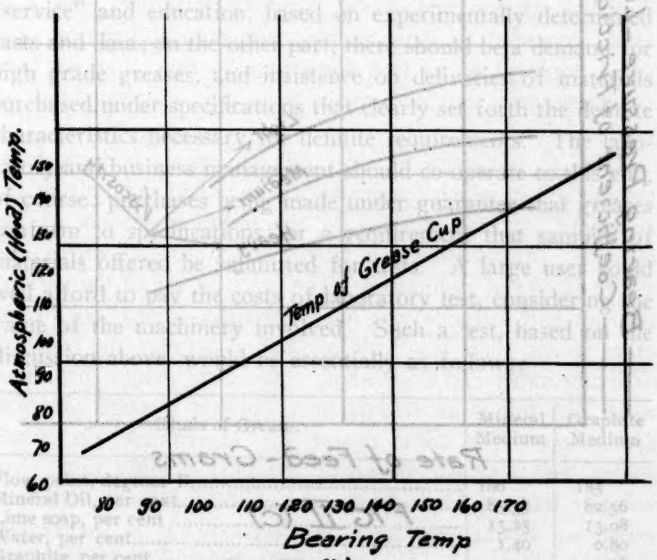


FIG. II (b)

Very many temperature, speed and pressure relations could be tabulated and plotted; such as those shown in Figure II. (Again, it is to be understood that these curves are illustrative only, and not based on actual figures.) For instance, taking greases of *standard* consistency and flow point, the rate of feed to the bearing could be determined. Fig. II (*a*).

Likewise, the resultant cup temperatures could be plotted, knowing the atmospheric (hood, Figure I) temperature, and the bearing temperature. This would be useful in furnishing information as to the proper *flow point* to prescribe for working cup-temperatures in actual practice. Fig. II (*b*).

A series of Curves, Fig. II (*c*), for each *viscosity of constituent oil* prescribed as standards, within the limits heretofore described, showing the coefficients of friction for varying rates of feed and bearing pressures, would indicate the proper oil viscosity to prescribe.

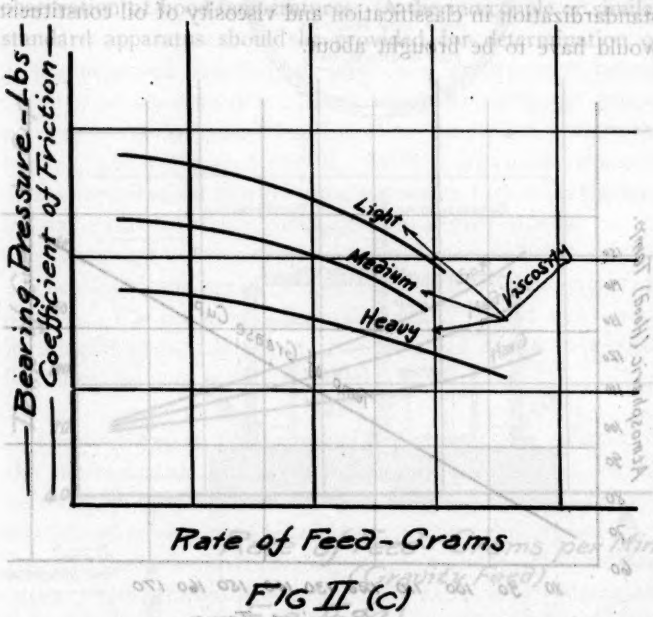


FIG II (c)

Many other interrelated factors could be similarly reduced to tabulated reference data.

III.

To an objection that placing grease on a scientific footing would result in an increase of costs, there are two answers;

First, that at *any* reasonable cost, lubrication is cheaper than repairs and tie-ups; Second, that a recent purchase of about 100,000 pounds of grease, under much more stringent specifications and test requirements than any hitherto used, both in respect to chemical quality of ingredients and viscosity of contained oil, was made at a price formerly quoted on a "grease is grease" basis, *i.e.*, consistency and soap content, chiefly.

The establishment of a rational practice is a matter of collaboration between manufacturers and larger users. As in the case of lubricating oils, there is on the one part a field for "service" and education, based on experimentally determined facts and data; on the other part, there should be a demand for high grade greases, and insistence on deliveries of materials purchased under specifications that clearly set forth the definite characteristics necessary for definite requirements. The laboratory and business management should co-operate to this end, of course; purchases being made under guarantee that greases conform to specifications; or a requirement that samples of materials offered be submitted for tests. A large user could well afford to pay the costs of laboratory test, considering the value of the machinery involved. Such a test, based on the discussion above, would be essentially as follows:

Kinds of Grease.	Mineral Medium	Graphite Medium
Flow point, degrees F	180	185
Mineral Oil, per cent	85.35	82.56
Lime soap, per cent	13.25	13.08
Water, per cent	1.40	0.80
Graphite, per cent	.....	3.56
Ash, as calcium sulphate, per cent	3.88	4.35
Consistence	20	16
Deleterious impurities	None	None
Chemical and physical properties of oil extracted from grease. Organic acidity as acid number	0.19	0.17
Saponification	None	None
Specific gravity at 60/60 degrees F	0.922	0.925
Flash point, open cup, degrees F	335	330
Fire point, open cup, degrees F	380	370
Saybolt Viscosity in seconds at 100 degrees F	162	203
Saybolt Viscosity in seconds at 130 degrees F	86	106
Saybolt Viscosity in seconds at 210 degrees F	40	43

CONVERGENTS AND THEIR IMPORTANCE IN PRACTICAL ENGINEERING.

By Karl F. Smith, Lieutenant Commander, U. S. Navy, Member.

Naval officers, and Naval Academy graduates, are generally conceded to have a fair mathematical training, and yet I think all of us find from time to time that we have forgotten very essential methods in the simplest kind of mathematics.

This, as it applies to older officers, comes perhaps from the method in which mathematics was taught not only at the Academy but also at most of the technical schools of the country. Mathematics was taught as an abstract science—as a sort of mental gymnastics. The practical use of the various theories and methods was not sufficiently emphasized or, in other words, there was little correlation between the teaching of mathematics and engineering. We were taught that there were such things as determinants, but not through practical application. We were taught the logarithmic decrement theory, but were not shown by practical example that what we were learning in the abstract was nothing more than that which we would meet later, in electricity, as a discharging curve of a condenser—or of damped oscillation in radio. We studied expansion of a series—but we were not told at the time, for example, that $x - \frac{x^3}{3!} + \frac{x^5}{5!} - \frac{x^7}{7!} \cdots$ gives us a method of working out the sine of any angle we may want to use, or in other words, gives us a method of working trigonometric problems without tables.

This applies not only to the higher branches of mathematics but to algebra or even to arithmetic.

Who would even think that inability to express a decimal as the nearest fraction would hold up temporarily repairs to an airplane, or the cutting of a gear? Yet this has actually happened to me. Plain algebra! and yet not a single person could help me out.

As this same problem has come up twice within a year on widely varied jobs, I am explaining this principle and its practical use for any others in the service who sooner or later may encounter the same problems. I claim no originality for rediscovering the method.

How many of us can, off hand, work out these three problems? Shop tables may contain some answers but not all.

1st. Cut a metric thread on an American lathe, with leading screw cut for "threads per inch."

2nd. Set a B. & S. milling machine to cut at an angle of 37 degrees 25 minutes 13 seconds.

3rd. Set the value π, or .62137 (the conversion factor from kilometers to statute miles) on the C or D slides of a slide rule so that the result is correct to 5 decimals.

I couldn't, I'm frank to say, and yet each of these problems involves simply the expression of a decimal as a fraction, or in other words, the "Theory of Convergents."

Similar problems arise for the engineer to solve at frequent intervals, and the method is the same in all.

EXPRESSING A DECIMAL AS ITS NEAREST PRACTICAL SIMPLE FRACTION.

Any number can be expressed as a simple continued fraction, terminable in the case of a commensurable quantity or non-terminating in the case of an incommensurable number.

The mechanical operation of finding this fraction is similar to finding the G. C. M. of two numbers. The number .3937, the conversion factor of metric to English units of length,

which occurs in the solution of the first example above, may be written $\frac{3937}{10000}$, and we proceed to determine the continued fraction as follows:

$$
\begin{array}{r}
3937)10000(2 \\
\underline{7874} \\
2126)3937(1 \\
\underline{2126} \\
1811)2126(1 \\
\underline{1811} \\
315)1811(5 \\
\underline{1575} \\
236)315(1 \\
\underline{236} \\
79)236(2 \\
\underline{158} \\
78)79(1 \\
\underline{78} \\
1)78(78
\end{array}
$$

and the continued fraction equal to $\frac{3937}{10000}$ is

$$
\cfrac{1}{2 + \cfrac{1}{1 + \cfrac{1}{1 + \cfrac{1}{5 + \cfrac{1}{1 + \cfrac{1}{2 + \cfrac{1}{1 + \frac{1}{78}}}}}}}}
$$

a terminable fraction.

I shall hereafter write the partial fractions similar to this,

$$(1)\ \frac{1}{2+},\ \frac{1}{1+},\ \frac{1}{1+},\ \frac{1}{5+},\ \frac{1}{1+},\ \frac{1}{2+},\ \frac{1}{1+},\ \frac{1}{78}$$

By simplification, we may write a simple fraction which will be approximately equal to this continued fraction, and hence, approximate the required decimal.

The convergents, as we call them, are successive approximations to this partial fraction.

The first convergent is the first term of this continued fraction or $\frac{1}{2}$, which is larger than .3937.

The second is $\dfrac{1}{2+\frac{1}{1}} = 1/3$, which is smaller than .3937

The third is $\dfrac{1}{2+\dfrac{1}{1+\frac{1}{1}}} = 2/5$, which is again too large.

Similarly the 4th convergent is 11/28 (too small), and the 5th convergent is 13/33 (too large), and the 6th convergent is 37/94 (too small), and the 7th convergent is 50/127 (too large).

It will be noted that to obtain the seventh convergent we have had to simplify a long continued fraction. This is simple, but tedious and a simple rule enables us to write the successive convergents directly.

QUICK RULE.

The numerator of the N^{th} convergent = numerator of (N-1) convergent times denominator of the N^{th} partial fraction, plus numerator of (N-2) convergent times numerator of the N^{th} partial fraction.

The denominator of the
N^{th} convergent = denominator of (N-1) convergent times denominator of N^{th} partial fraction, plus denominator of (N-2) convergent times numerator of the N^{th} partial fraction.

In the above case it is desired to write down the 7th convergent, after working the 5th and 6th. Let us see what we have. We want the seventh convergent or N = 7.

The (N-2) convergent—or the fifth previously worked out is 13/33.

Let A = The numerator of (N-2) convergent 13, and
Let B = The denominator of (N-2) convergent 33.

Similarly the (N-1) convergent or sixth is 37/94.

Let C = The numerator of the (N-1) convergent 37, and

Let D = The denominator of the (N-1) convergent 94.

The next partial fraction (*i.e.*, the seventh) to be added is seen from (1), to be $\frac{1}{1}$.

E = The numerator of the next fraction (the seventh).

F = The denominator of the next fraction (the seventh).

Now applying the rules given

Numerator of N = (C × F) + (A × E), and the
denominator of N = (D × F) + (B × E)

or

$$N = \frac{(C \times F) + (A \times E)}{(D \times F) + (B \times E)} = \frac{(37 \times 1) + (13 \times 1)}{(94 \times 1) + (33 \times 1)} = \frac{50}{127}$$

This is the seventh convergent.

Similarly we write the eighth =

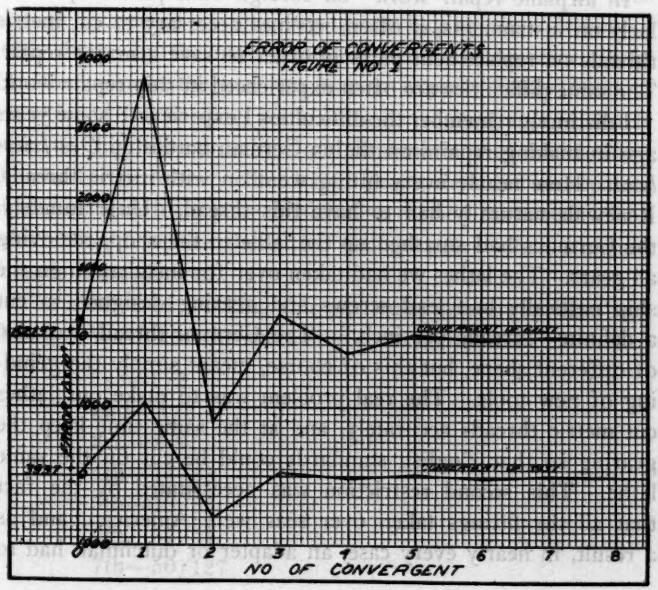
$$\frac{(50 \times 78) + (37 \times 1)}{(127 \times 78) + (94 \times 1)} = \frac{3900 + 37}{9906 + 94} = \frac{3937}{10000}$$

Tabulating the values of the convergents and the errors of each from the true value, we get:

TABLE NO. 1.

No. of Convergent	Computed Value of Convergent			True Value of Convergent	$\Delta \times 10^4$	$\Delta \times 10^4$ Odd	$\Delta \times 10^4$ Even
1	1/2	=	.5000	.3937	+ 1063	+ 1063	
2	1/3	=	.3333		− 604		− 604
3	2/5	=	.4000		+ 63	+ 63	
4	11/28	=	.3929		− 8		− 8
5	13/33	=	.3939		+ 2	+ 2	
6	37/94	=	.3936		− 1		− 1
7	50/127	=	.393701		+ .01	+ .01	
8	3937/10000	=	.3937		+ 0		− 0

A curve, using number of convergent and Δ from table No. 1 as co-ordinates, is shown in figure 1.



This curve shows us certain interesting and instructive facts about these convergents, which is true of all:

1st. Every convergent is in its lowest terms.

2nd. The odd convergents form a decreasing series of rational fractions whose value approaches that of the whole convergent fraction. The same is true for the even convergents, but in this case the series is increasing.

3rd. Every odd convergent is less than, and every even convergent is greater than any following convergent.

4th. Each successive convergent is nearer the true value of the whole fraction than the one preceding it.

5th. Every convergent is a closer approximation to the real value of the fraction than any other fraction whose denominator is smaller than that of the convergent.

THE PRACTICAL USE OF CONVERGENTS.

In airplane repair work—on foreign built planes—particularly on instruments, it often becomes necessary to cut metric threads on replacement parts in order to save a major repair. The same difficulties are often experienced in ship repair work.

I distinctly remember the difficulties I experienced some years ago in making repairs on the ex-German battleship *Oestfriesland*. Here nearly every fitting went on with metric threads. Renewals naturally had to have this thread. Unfortunately, the Germans had smashed up the lathe in the shop, removing all gears. The lathes we had taken over for installation on the ship naturally had lead screws and gearing designed to cut a definite "threads per inch." The German thread was also of different shape, but this of course, meant simply the grinding of new tools. The real problem was to make our lathes cut metric threads accurately, and at the same time keep the gearing so that English standards could be cut on the same lathe. This second restriction was necessitated by the fact that all the fittings taken over seas were American, and as a result, in nearly every case, an adapter or dutchman had to

be constructed having bastard threads, metric on one end and English on the other.

The problem boiled down to this; Desired, that with the gear box setting set to cut N threads per inch the lathe would actually cut N threads per centimeter. This meant installing between the back gearing and the lead screw shaft gear, a pair of gears (one an idler to keep motion in original direction) having a reduction ratio equal to the conversion factor for centimeters to inches—or in other words, a conversion factor ratio of .3937. That is, the number of teeth on the first gear, to the number on the gear driving the lead screw must be in the ratio of 3937 to 10000. The intermediate gearing was a 1 to 1 idler on the metric gear. The problem reduced to this—cut two gears in such a manner that the number of teeth on the first to the number on the second shall be in the ratio of 3937/10000, remembering that this ratio must be expressed as a simple fraction with numerator and denominator of such size as to permit cutting two gears having the number of teeth shown by the numerator and denominator respectively. In other words, the problem was to express 3937/10000 as a continued fraction and work out a convergent approximately equal to this, the numerator and denominator of which show the number of teeth on the change gears, mounted on separate shafts on a shifting plate, which also carried two equal idlers, which would replace these two change gears when standard threads were desired.

The method of working this particular convergent has been shown: We found the successive convergents to be:

1st—1/2.
2nd—1/3.
3rd—2/5.
4th—11/28.
5th—13/33.
6th—37/94.
7th—50/127.

Studying these, we see either the sixth or seventh would give us practical gears. The seventh would be selected in obedience to the fourth statement above, "Each successive convergent is nearer the true value of the whole fraction than the one preceding it." Now, consider the accuracy of this approximation, 50/127 works out to be .393701.

The error, therefore, in per cent is

$$\frac{.393701 - .3937}{.3937} \times 100,$$

or about 1/4000 of 1 per cent, which is too slight to be detected except on a measuring machine.

The actual cutting of the gears presented another problem: That of setting the dividing head of the gear cutter for an unusual angle. The method to be used in cases of this kind being shown in the next section.

SETTING THE DIVIDING HEAD OF A MILLING MACHINE FOR AN UNUSUAL ANGLE.

This second problem was again brought to my attention in inspecting an airplane job. Computations showed that it would be necessary to set an angle of 37 degrees, 25 minutes, 13 seconds on a B. & S. milling machine. Three dial plates were available with circles shown below:

1st—15, 16, 17, 18, 19, 20.
2nd—21, 23, 27, 29, 31, 33.
3rd—37, 39, 41, 47, 49.

What plate should be used and what setting on that plate? The machine to be used was equipped with the ordinary type dividing head, forty turns of the crank giving one turn to the spindle. From this it is seen that one turn of the crank would give 360/40 or 9 degrees. The angle to be cut is 37 degrees 25 minutes 13 seconds. Four turns of the crank would give 36

degrees, leaving 1 degree 25 minutes 13 seconds to be turned on the dial plate. This equals 5113 seconds. One complete revolution is $9 \times 60 \times 60 = 32400$ seconds. Our problem is then to find a continued fraction equal to 5113/32400 and from this to find a convergent whose denominator lies between 15 and 49, the limits forced by the dial plates available. Working this fraction, as before, we find it to be:

$$\frac{1}{6+} \quad \frac{1}{2+} \quad \frac{1}{1+} \quad \frac{1}{31+} \quad \frac{1}{2+} \quad \frac{1}{26+}$$

The convergents are:

1st = 1/6.
2nd = 2/3.
3rd = 3/19.
4th = 95/602.

The convergent, according to rule five previously given, which best fits, the denominator being between 15 and 49, is 3/19, the third convergent. Our setting, therefore, becomes four full turns of the crank and three holes in the 19 hole circle.

$3/19 = .15790.$

The error of this setting is

$$\frac{.15790 - .15780}{.15780} \times 100 = .0634 \text{ per cent.}$$

SLIDE RULE SETTINGS.

Last, but probably most important of all, due to the frequency of their use, is the question of proper slide rule settings.

Anyone who uses a slide rule cannot fail to realize how hard it is to set the value of π on the C or D scales with any degree of accuracy, due to the fact that the eye cannot accurately set off 142/1000 of the small spaces between graduations on the slide rule. The inability to do this gives a very appreciable

error in any computation involving the value of π. Hundreds of other constants give similar errors, and where these constants are used frequently, it certainly pays to work out and remember an accurate slide rule setting.

This, again, involves the determination of a continued fraction equal to the constant and a practical convergent approximately this value, that is to say, one having a numerator and denominator, both of which can be set with accuracy on the C and D scales. Thus, $\pi = 3.14159 = \frac{314159}{100,000}$, which, worked out, gives the following partial fractions:—

$$3 + \frac{1}{7+} \quad \frac{1}{15+} \quad \frac{1}{1+} \quad \frac{1}{25+} \quad \ldots\ldots$$

The first two convergents are written by inspection and the others computed by the rule and are:

$$1\text{st} = 3$$

$$2\text{nd} = \frac{22}{7}$$

$$3\text{rd} = \frac{(22 \times 15) + (3 \times 1)}{(7 \times 15) + (1 \times 1)} = \frac{333}{106}$$

$$4\text{th} = \frac{(333 \times 1) + (22 \times 1)}{(106 \times 1) + (7 \times 1)} = \frac{355}{113}$$

$$5\text{th} = \frac{(355 \times 25) + (333 \times 1)}{(113 \times 25) + (106 \times 1)} = \frac{9208}{2931}$$

Either the third or fourth convergent furnishes an excellent setting for π on a slide rule. The fifth, due to the four places is as difficult to set as the original value 3.14159, direct. To use practically, set 355 over 113 and under 1 find π, or set 333 over 106 and under 1 find π. In the first case the actual value is 3.141593 or an error in the sixth decimal place.

As a final example let us obtain a slide rule setting for the conversion of kilometers per hour to miles per hour. This factor is .62137. Here the partial fractions are:

$$\frac{1}{1+}\ \frac{1}{1+}\ \frac{1}{1+}\ \frac{1}{1+}\ \frac{1}{1+}\ \frac{1}{3+}\ \frac{1}{1+}\ \frac{1}{2+}\ \ldots\ .$$

The convergents by rule are:

$$\frac{1}{1},\ \frac{1}{2},\ \frac{2}{3},\ \frac{3}{5},\ \frac{5}{8},\ \frac{18}{29},\ \frac{23}{37},\ \frac{64}{103}$$

The actual value of $64/103 = .62136$ so that setting 64 over 103, to convert kilometers to miles introduces an error of the slide rule computations of one part in 62,000, an error plainly negligible.

Whenever an engineer uses certain constants frequently, it certainly pays to work out the convergent for such accuracy as he may desire, and paste the constant on the back of the rule.

In this connection remember that the odd numbered convergents are always larger or equal to the true value while the even convergents are smaller or equal to the true value. This has an important bearing on the limit and direction of errors desired.

Hint:—

To remember the method of computation.

If the (N-2) convergent is written first, the numerator being lettered A and the denominator B, then the (N-1) convergent written, and its numerator lettered C and denominator D, and finally the numerator and denominator of the N^{th} partial fraction called E and F respectively.

"Can't find any error" is the key.

$$N = \frac{\text{Can't find any error}}{(C \times F) + (A \times E)}{(D \times F) + (B \times E)}$$

Remembering at the same time that F and E appear in the same order in both numerator and denominator, and that other letters follow in the sequence they appear in the alphabet.

DIESEL ENGINES IN SUBMARINES.*

By E. C. MAGDEBURGER,

BUREAU OF ENGINEERING, NAVY DEPARTMENT.

The submarine and the Diesel engine came out of the experimental stage about the same time. Twenty years ago various navies of the world began to acquire submarines, thus recognizing their usefulness as a warship. Although their size in those days was extremely modest, the functions required were essentially the same. The boat had to have high speed on the surface, considerable capacity for submerged operation, had to carry a reasonable number of torpedoes and provide quarters—not living quarters by any means, but elbow room—for the necessary personnel to operate such a mass of diversified and complicated machinery. The first submarines were equipped with gasoline engines and the United States Navy apparently was the only one that made a boat so equipped a reasonable success, despite the dangers from explosions and the so-called gasoline-jags. European nations, being better acquainted with the Diesel engine, recognized the possibilities of this type of prime mover sooner, and in 1908 the author had the pleasure of participating in the discussion of the feasibility of designing and building Diesel engines to replace gasoline engines in the Russian submarines, built in Germany.

The task was a mean one, especially because so much was expected at once, and it took the Diesel engine builder a

---

* Paper read before a joint meeting of the Engineering Societies of Washington, D. C., during "Oil and Gas Power Week," April 23, 1925.

FIGURE 1.

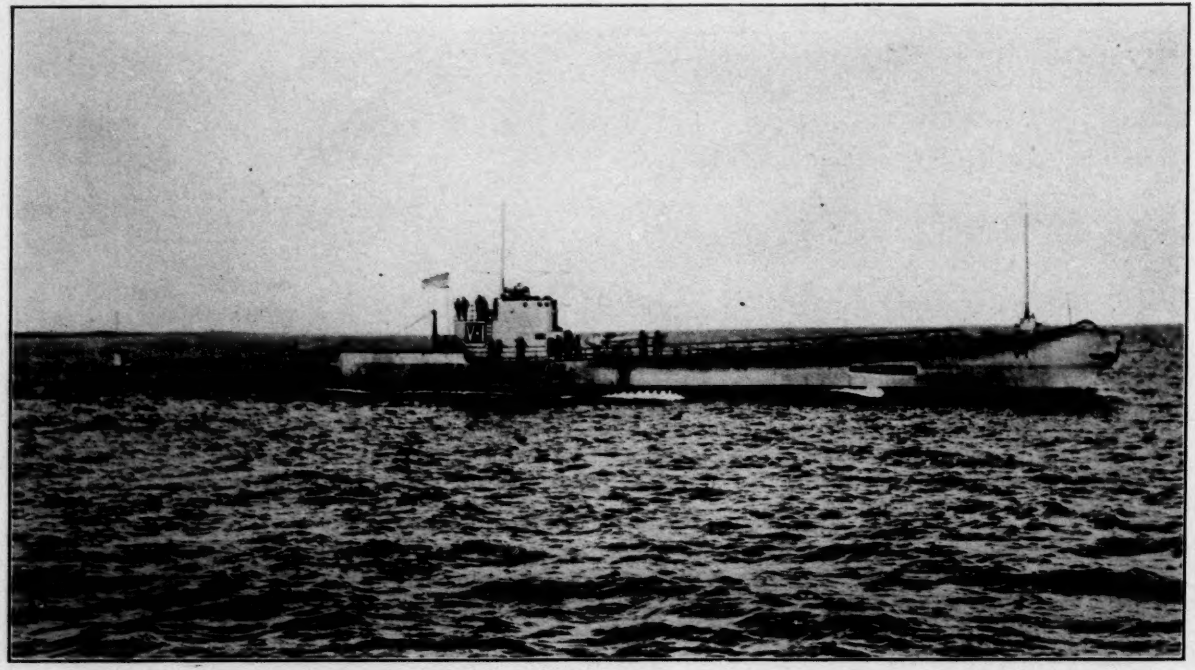


FIGURE 2.

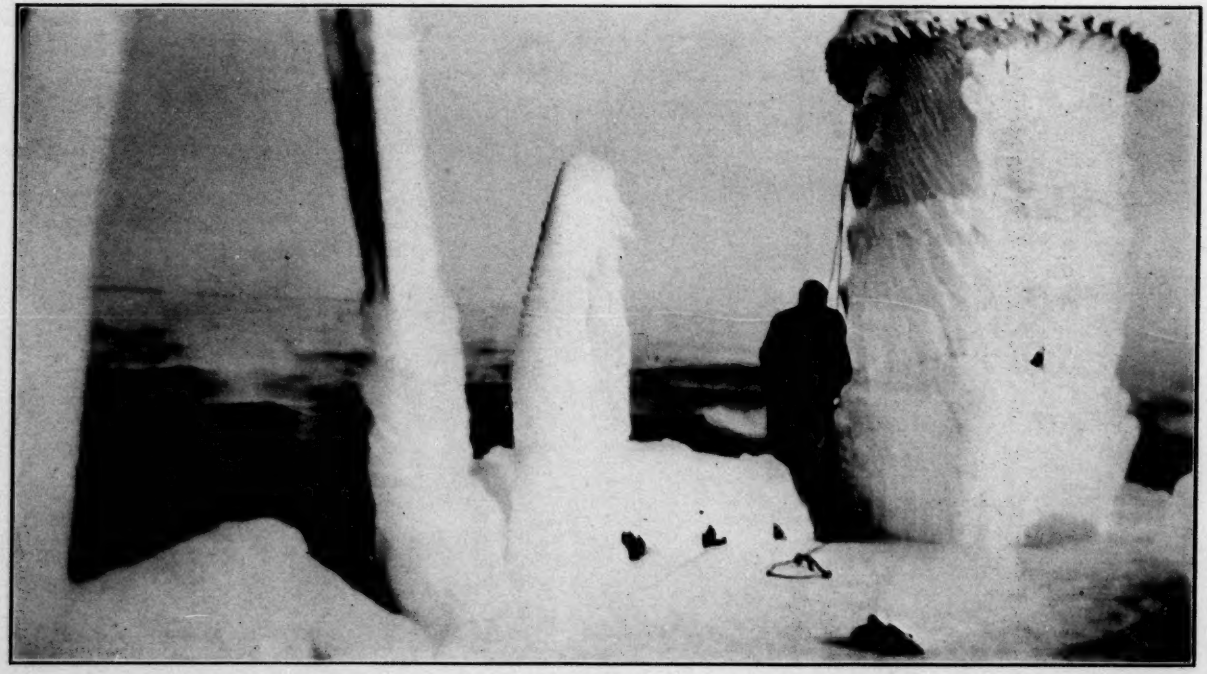


FIGURE 3.

number of years before the point of view of the naval officer was sufficiently modified to give the Diesel engine a better chance. Meanwhile attempts were made to design and build an engine that would comply with the extremely severe requirements of weight and space insisted upon.

Any engineer realizes that the specific weight of any kind of apparatus per unit of output is much larger in smaller sizes than in medium and large sizes. Besides, the size of the operator remained the same, whereas the unobstructed diameter of the inner shell grew from about 13 feet 6 inches in the first U. S. Submarine to receive Diesel engines, to about 20 feet in the modern fleet submarine. The relative size of these two boats can be seen from the accompanying photographs. Figure 1 shows one of the first two United States submarines equipped with Diesel engines, *E-1* and *E-2*, and Figure 2 the modern submersible Leviathan—*V-1*.

To illustrate the difficult conditions under which the operating personnel of submarines have to work and the consequent necessity for space to be devoted to their reasonable comfort, I cannot refrain from showing a most striking picture of one of the modern boats taken en route to Portsmouth, N. H., in the winter of 1922.—Figure 3.

Before proceeding with the main part of the paper, permit me to refresh your memory with a brief description of essential events occurring in the cylinder of a Diesel engine and all the elements that are necessary for it.

The term Diesel engine means an internal combustion engine that can burn directly in the cylinder heavy liquid fuels of high boiling point, using the heat developed by the compression of the air charge to ignite the fuel injected into the cylinder at or near the end of the compression stroke.

The simplest to understand is the so-called four-cycle engine using four strokes of the piston, or two revolutions of the crank-shaft, for the complete cycle of events. Figure 4, shows schematically the essential parts of such an engine.

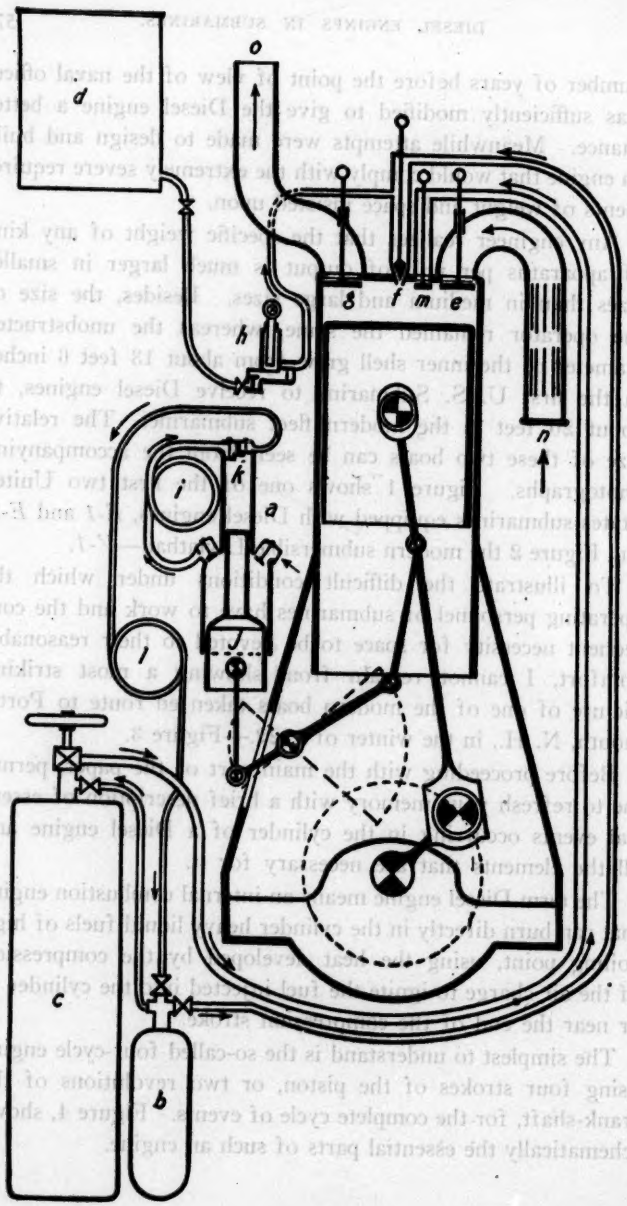


FIGURE 4.

The engine piston has started on its downward travel. The air-admission valve *e* is open to the atmosphere and lets air into the engine cylinder. On its return stroke the valve *e* and all other valves are closed, so that the piston compresses the air in the cylinder to a pressure of 450 to 500 pounds per square inch. As a result the air becomes highly heated, its temperature rising to about 1,000 degrees F. when the highest pressure is reached. When the piston is at its upper dead center, the fuel-injection valve *f* is opened and liquid fuel in a volume proportioned to the load of the engine is forced into the cylinder, where, meeting the highly heated air, it is automatically ignited, and burns. When a certain amount of fuel has been delivered, valve *f* closes. Under the impulse of the expanding gases, the piston moves downward, transforming the heat energy of the fuel into work. Arriving at its lower dead center, the piston reverses its travel and begins the upward stroke. The exhaust valve *g* being open, the piston sweeps the products of combustion before it, expelling them through this valve into the atmosphere, thus completing the four-stroke cycle. The cycle is repeated, when air valve *e* opens again, and the piston starts on its downward stroke.

In order to better visualize this cycle and also for purposes of comparison of the four-stroke and two-stroke cycles, a series of diagrammatic views showing the relative position of piston and valves was made. Figure 5 shows the four cycle events as follows:

(*a*) First stroke (admission stroke). Piston travels down; admission valve open; cylinder is being filled with air.

(*b*) Second stroke (compression stroke). Piston travels up; all valves closed; air in cylinder is being compressed.

(*c*) Third stroke (power stroke). Piston travels down; fuel valve open at top dead-center, but closed at fraction of stroke; gases expand.

(*d*) Fourth stroke (exhaust stroke). Piston travels up; exhaust valve open; burnt gases are expelled from cylinder.

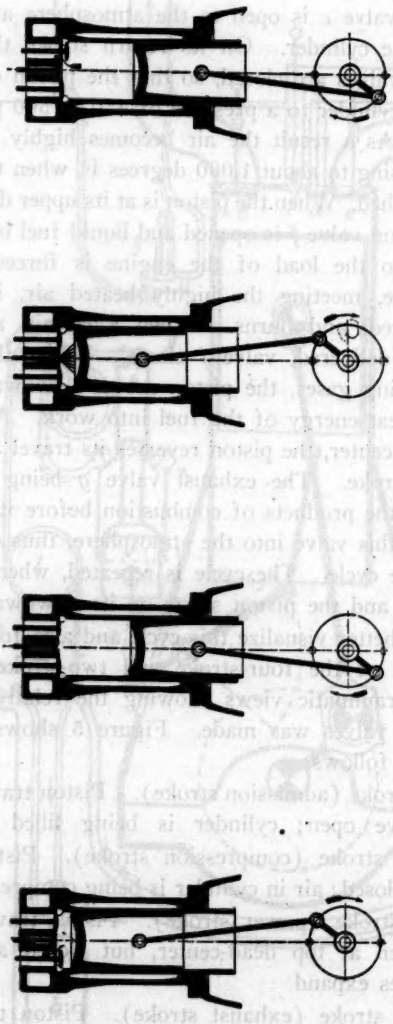


FIGURE 5.

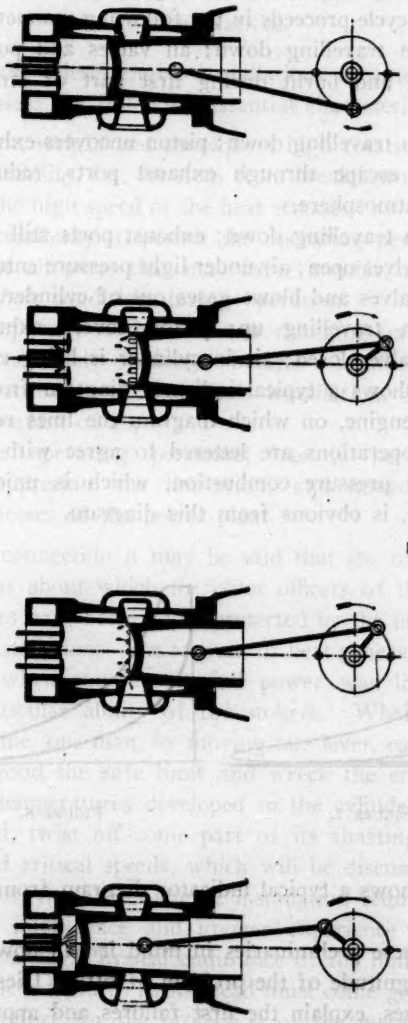


FIGURE 6.

Figure 6 shows one of the existing types of two-cycle engine and here the cycle proceeds in the following manner:

(*a*) Piston travelling down; all valves and ports closed; fuel injected and burnt during first part of stroke; gases expand.

(*b*) Piston travelling down; piston uncovers exhaust ports; burnt gases escape through exhaust ports, reducing their pressure to atmosphere.

(*c*) Piston travelling down; exhaust ports still uncovered; scavenging valves open; air under light pressure enters through scavenging valves and blows gases out of cylinder.

(*d*) Piston travelling up; piston covers exhaust ports, scavenging valve closed; air in cylinder is being compressed.

Figure 7 shows a typical indicator diagram from a four-cycle Diesel engine, on which diagram the lines representing the various operations are lettered to agree with Figure 5. The constant pressure combustion, which is unique in the Diesel engine, is obvious from this diagram.

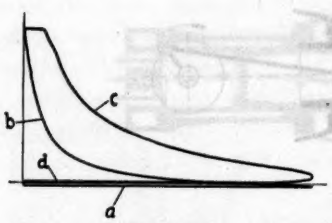


FIGURE 7.

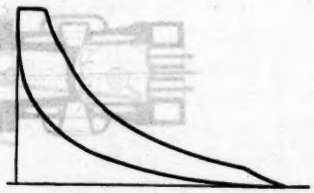


FIGURE 8.

Figure 8 shows a typical indicator diagram from two-cycle Diesel engine.

With all these preliminaries in mind it will now be easier to see the magnitude of the problem of fitting Diesel engines into submarines, explain the first failures and appreciate the success of modern installations.

ESSENTIAL REQUIREMENTS FOR DIESEL ENGINES IN SUBMARINES.

Before the Diesel engine installations became a success it had to possess the following essential characteristics:

(*a*) Light weight and consequent high rotative speed, with
(*b*) Reliability, *i.e.*, freedom from breakdown due either to the high speed or the heat stresses.
(*c*) Accessibility to permit the operating personnel to get at all parts that may cause trouble, and to watch their condition during operation.
(*d*) Simplicity, so that the ever changing naval crews can reasonably quickly become familiar with the intricacies of the whole installation, and most important of all,
(*e*) The operating personnel has to be educated to appreciate the limitations and unavoidable weaknesses of the power plant.

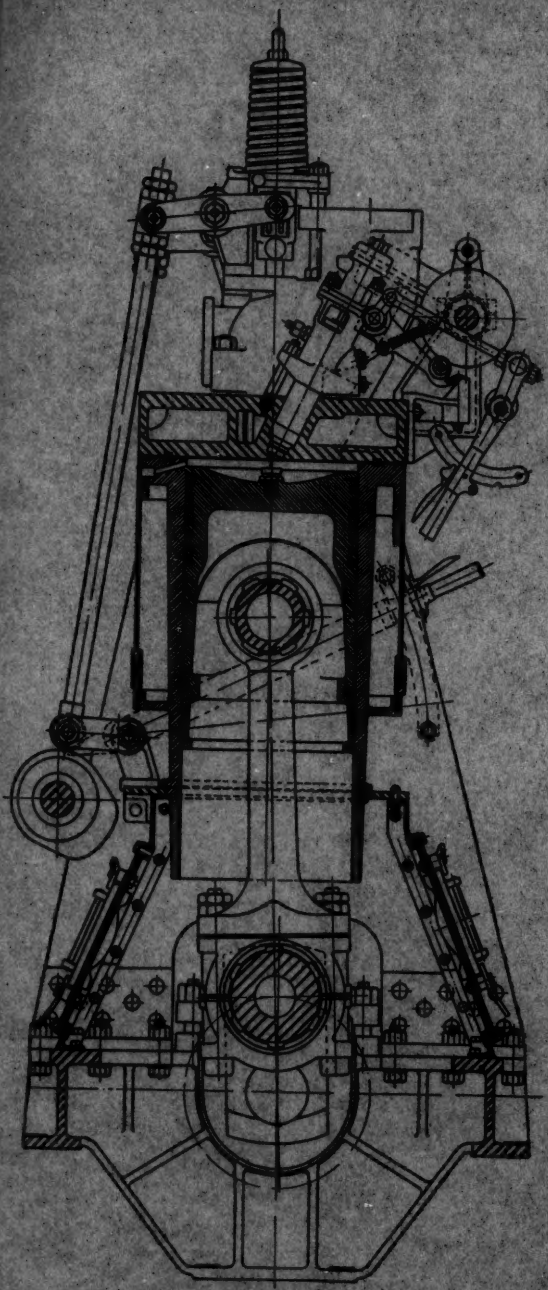
In this connection it may be said that the old-time steam installations about which the older officers of the Navy sing high praises have been amply protected by the human element of these installations—the amount of heat generated and steam produced when running at full power was limited mainly by the muscular ability of the stokers. Whereas with the Diesel engine, one man, by moving one lever, can load up the engine beyond the safe limit and wreck the engine through excessive temperatures developed in the cylinders; or on the other hand, twist off some part of its shafting by running at so-called critical speeds, which will be discussed later. In other words, the Diesel engine installation requires a greater degree of intelligence and greater experience of the operating personnel. Physical endurance of the human being has ceased to be a factor. In its stead must come good judgment, conscious effort for the protection of the installation, and a thorough knowledge of all the forces that may be set up in

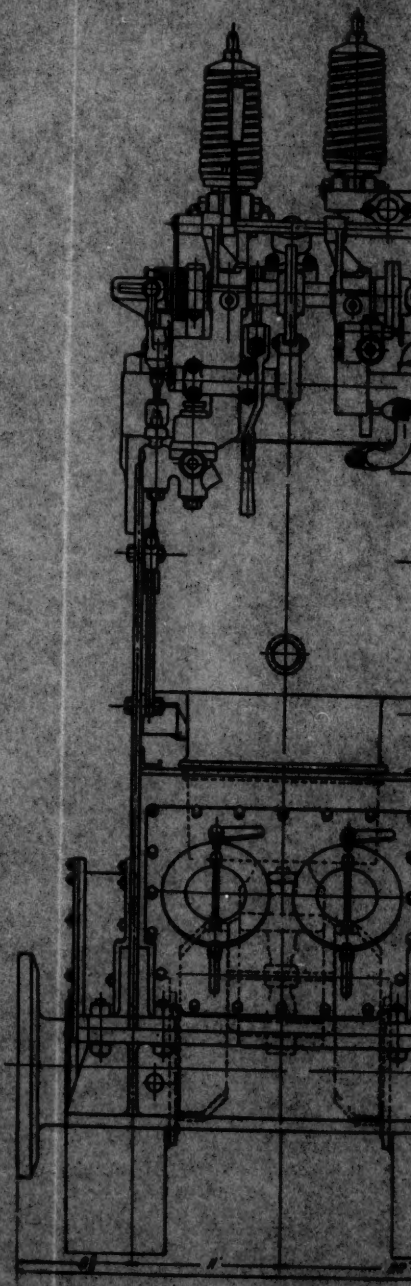
the parts of the engine by any action or lack of attention by the engineer on watch. Though many details of the Diesel engine are made fool-proof the same cannot be said of it as a whole, and a watchful eye, a sharp ear and quick brain are among the prime requirements of Diesel engine operators in submarines.

FIRST DIESEL ENGINES INSTALLED IN SUBMARINES.

First Diesel engines to be installed in United States submarines were of the Vickers air injection type, 4-cycle, 4-cylinder, non-air starting and non-reversing units. They were built by the New London Ship and Engine Company and were installed in Submarines *E-1* and *E-2*, commissioned February, 1912. Six cylinder engines of similar cylinder dimensions were installed in the "F" Class of submarines commissioned in the summer of 1912. They were 12¾-inches bore with 13½-inches stroke and developed 275 and 410 B.H.P. respectively at 400 R.P.M. Figure 9 is the general arrangement of the 4-cylinder engine. The cylinders were mounted on steel plate columns, secured to cross girders of bedplate. The camshaft for operating the inlet and exhaust valves was on the outboard side of the engine, mounted in brackets secured to the columns. The shaft was driven by spur gears located in the middle of the engine and from it also were driven the circulating water pump and main fuel and lubricating pumps. On the inboard side of the engine was another shaft driving the fuel valves and fuel measuring pumps. This shaft was driven from main cam shaft by two sets of spiral gears through an inclined intermediate shaft also in the center of the engine.

The bedplate was made of cast steel in two sections bolted together. Main bearings had composition shells lined with white metal and fitted with water circulation. Cylinders were a simple cylindrical casting fastened to the cylinder head and fitted with separate cast iron jackets shrunk in place.





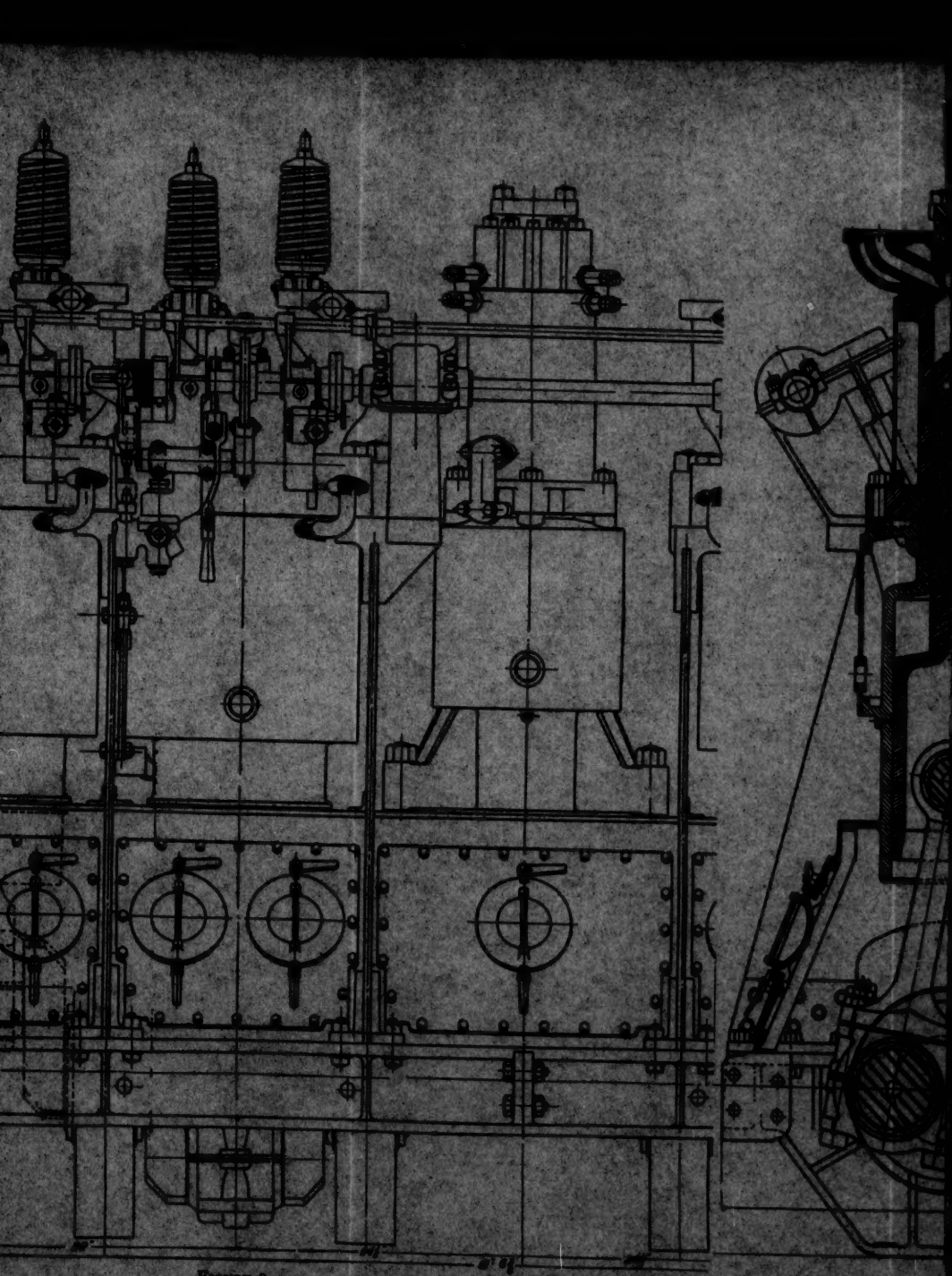


FIGURE 9.

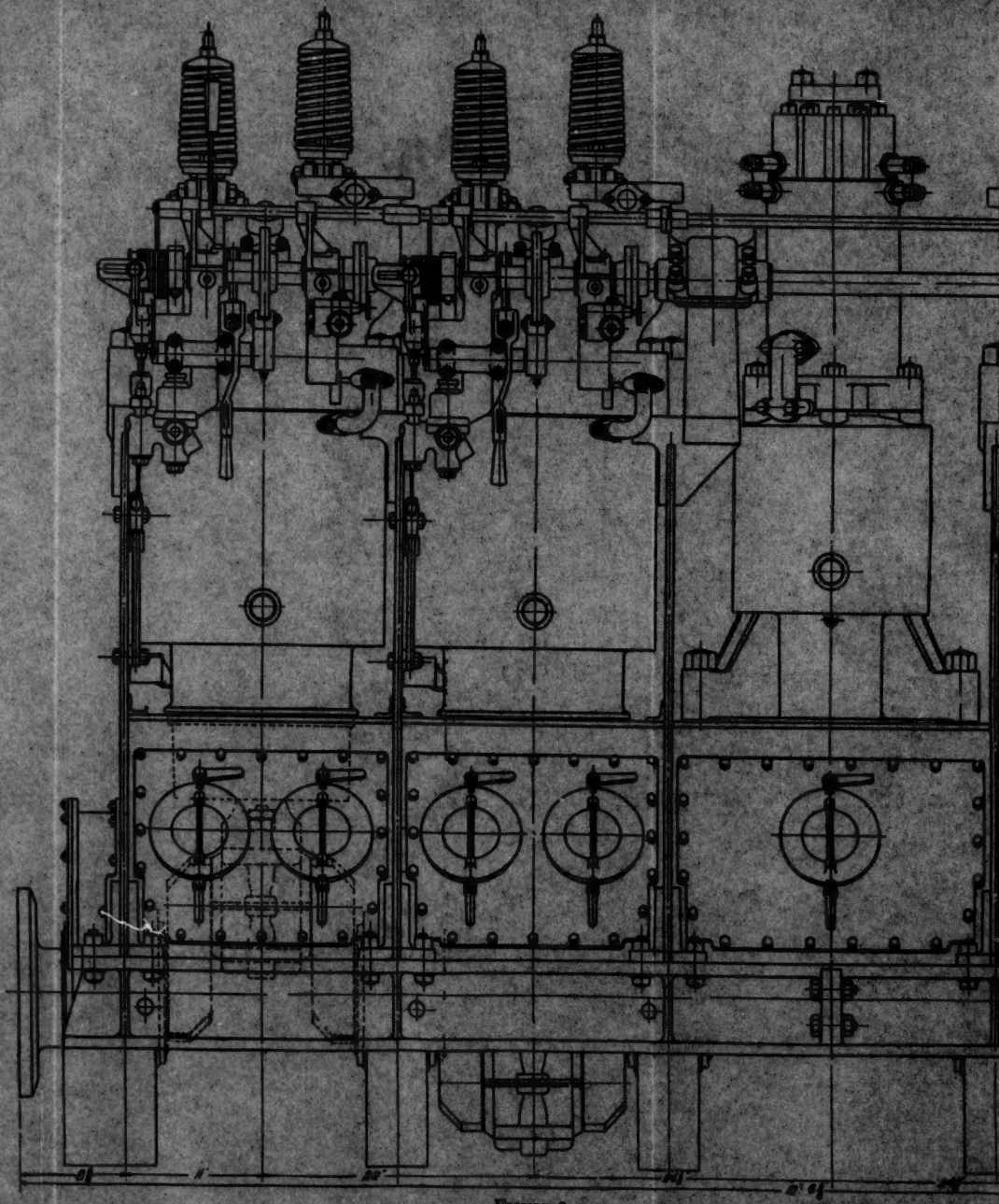
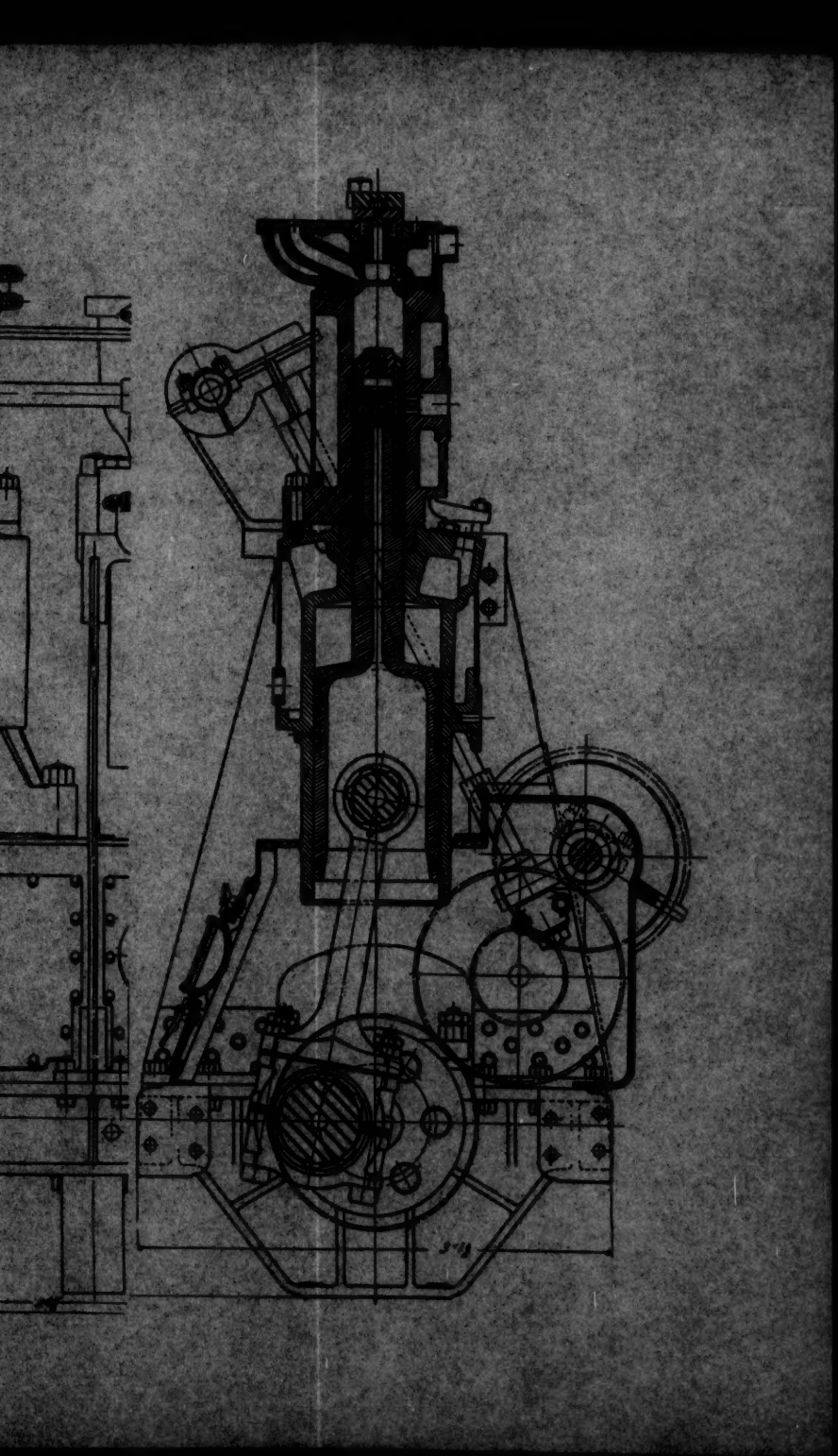


FIGURE 9.



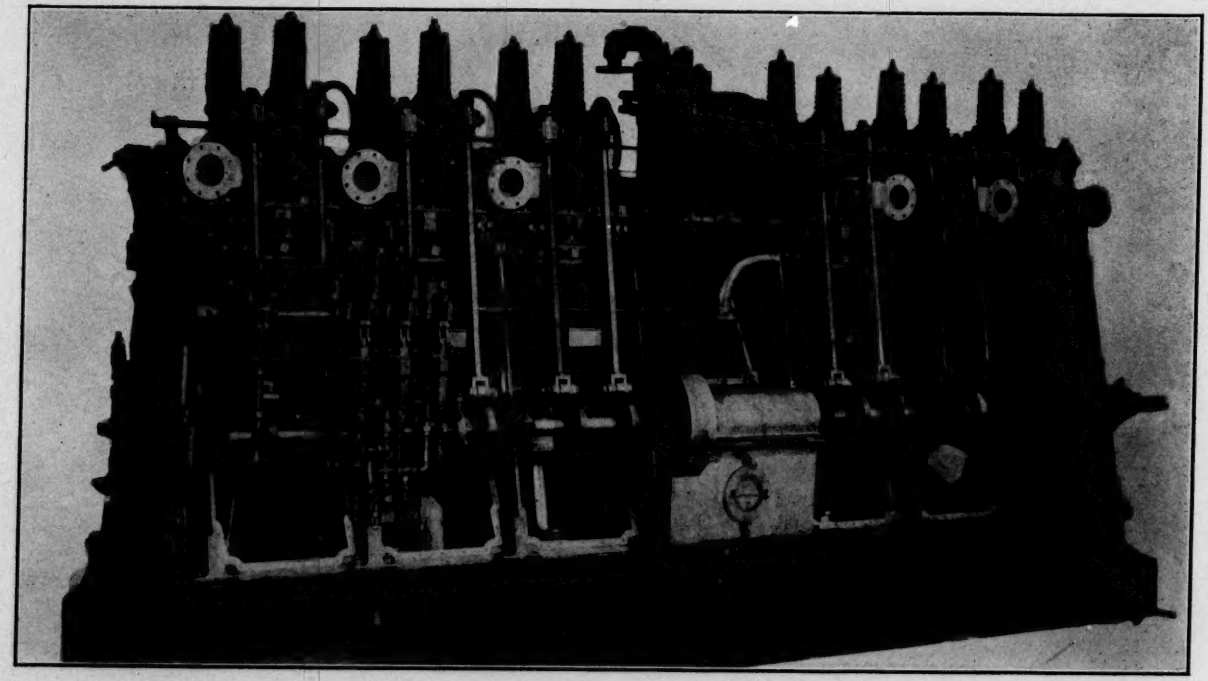


FIGURE 10.

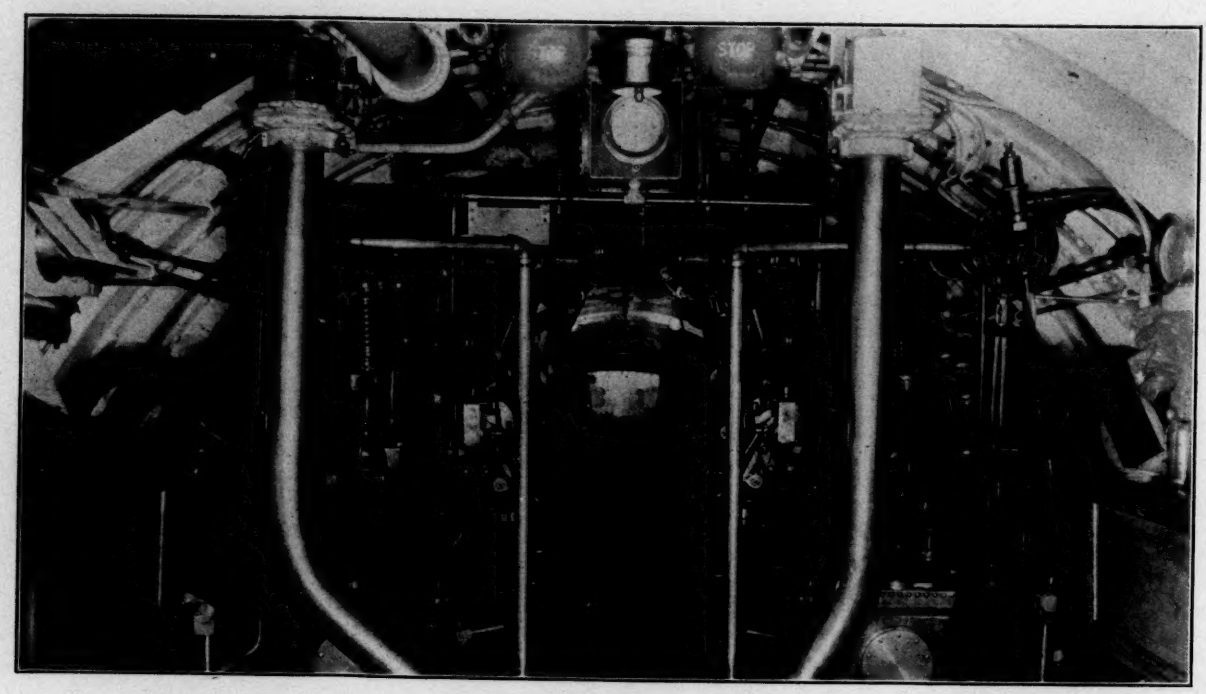


FIGURE 11.

Cylinder heads were a comparatively simple steel casting with openings for the inlet and exhaust and fuel valves. A cast iron ring ground in place served to make the joint between the cylinder and head.

Crankshaft consisted of two sections for working cylinders with a one throw crank between them for the air compressor. Both the inlet and exhaust valve cages had the pipe flange connecting to the respective headers cast integral with the cage. The exhaust valve cage was water cooled and so was the exhaust valve. Each cylinder had its own fuel measuring pump driven by the inboard shaft on the cylinder head. The quantity of fuel pumped was regulated by a mechanically controlled suction valve by means of a stepped cam mounted on fuel pump shaft and capable of lateral motion so that the valve could be operated by the various steps of this cam.

Two-stage air compressor was located in the middle of the engine and secured to brackets bolted to the two center columns. To obviate the removal of high pressure cylinder for inspection of piston rings, these were mounted on a sleeve secured by a stud and could be removed by merely removing the high pressure cylinder head. Only discharge valves were fitted to both stages, ports in the cylinder walls serving for the inlet of air.

Water pump of the rotary vane type was driven from aft end of cam shaft.

Forced feed lubrication was used but the crankshaft was not filled the entire length, each main bearing supplied oil to one crank and wrist pin of the cylinder next to it, thus permitting individual regulation.

Figure 10 shows the outboard view of the six cylinder engine installed in "F" Class of submarines and Figure 11 is a view of the engine room looking aft.

Only points of interest were taken from the description of these engines to illustrate the features which caused the most trouble to the operating personnel; these troubles were as

follows: (*a*) air compressor valves would break and drop into cylinders, causing its lugs to give way, rupture piping and cause havoc in the engine room; (*b*) water cooled exhaust valves and valve cages, caused lots of grief through cracking—such complicated castings are even now a mean thing to make; (*c*) water cooled main bearings are a thing of the past now because in service the water joints in the bedplate would leak and salt up the lubricating oil, destroying its lubricating properties; (*d*) cylinder studs gave continuous trouble—the main reason being that some of the studs were installed between the two cylinder heads and their nuts in service would sometimes rest only on one head and cause the other studs to snap. Such practice is permissible only when the two elements are held rigidly together. These are the principal objections of the operating personnel, but examination of Figure 11 and consideration of general conditions prevailing in 1911 and 1912 finish the story of their doom. Built too light, installed in too small a space with some of their principal auxiliaries and cam shaft drive outboard, and no room for anybody to get at them, a new departure for most people that had to operate them as well as to their builders, these engines were finally taken out and replaced by more reliable engines.

TWO CYCLE DIESEL ENGINES.

Meanwhile there were two fundamentally different types of two-cycle engines developed in continental Europe and engineering opinion, including Dr. Diesel, the inventor himself, seemed to favor the two-stroke cycle for marine propulsion. At any rate the first Diesel engine built in Germany* for submarine service was an 850 B.H.P. 2-cycle, built in 1911 by Krupp for the Italian Navy. This engine was of a valve scavenging type with a separate scavenging pump. At the same time Sulzer Brothers developed a port scavenging type of engine, a number of which were also furnished to

* See Der Bau von Unterseebooten auf der Germanawerft by H. Teckel, published by V. D. I. 1923.

the Italian Navy. Maschinenfabrik Augsburg–Nurnberg, familiarly known as M.A.N. also developed a two-cycle submarine engine of the valve scavenging type, but using a step piston for the supply of the scavenging air.

One big reason for the favor of the two-cycle engine for marine service was the simplicity of mechanism necessary to make it reversible since no inlet and exhaust valves had to be taken care of.

The New London Ship and Engine Co., Nelseco, of Groton, Conn., purchased the license from the M.A.N. Co., and the next Diesel engines built by this company were of the M.A.N. two-cycle valve scavenging type installed in the "H" and "K" Classes of submarines commissioned December 1913–December 1914.

M.A.N.—NELSECO TYPE.†

These engines had 6 working cylinders of 11-inches bore, 12¼-inches stroke and were designed to develop 450 B.H.P. at 450 R.P.M.

Figure 12 shows the cross-section of the working cylinder and the details of its construction. Figure 13 shows the engine on the test block.

The bedplate was of Vanadium bronze, cast in two sections, bolted together. The lower shells of main bearings were also of vanadium bronze and arranged for oil cooling. The bearing caps were iron castings and both were white metal lined. The crankshaft was of nickel steel, made in two sections. The camshaft was arranged above the cylinder tops and driven by means of two sets of spiral gears through an intermediate vertical shaft from the aft end of the engine. The air compressor was driven by a separate crank on the forward end of the crankshaft which was also intended to mount an eccentric to drive the fuel measuring pump and a spur gear for driving the circulating water and lubricating oil pumps.

† For more complete description of engine and tests see JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, 1913, p. 447.

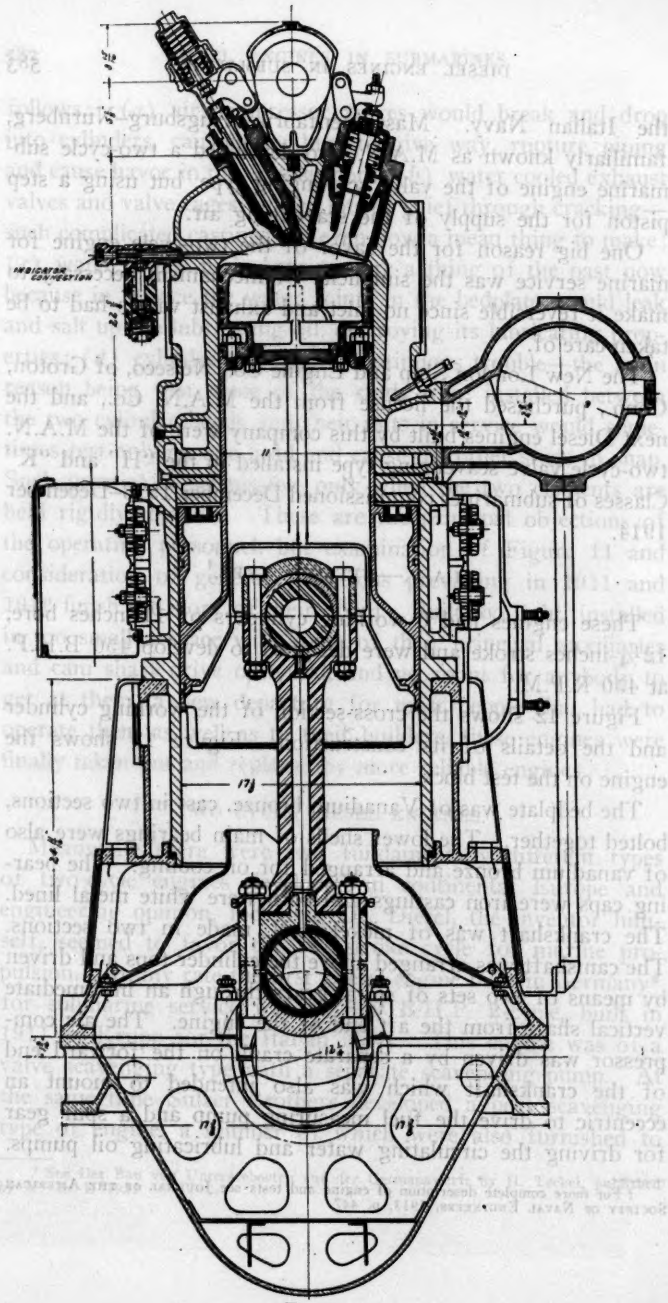


FIGURE 12.

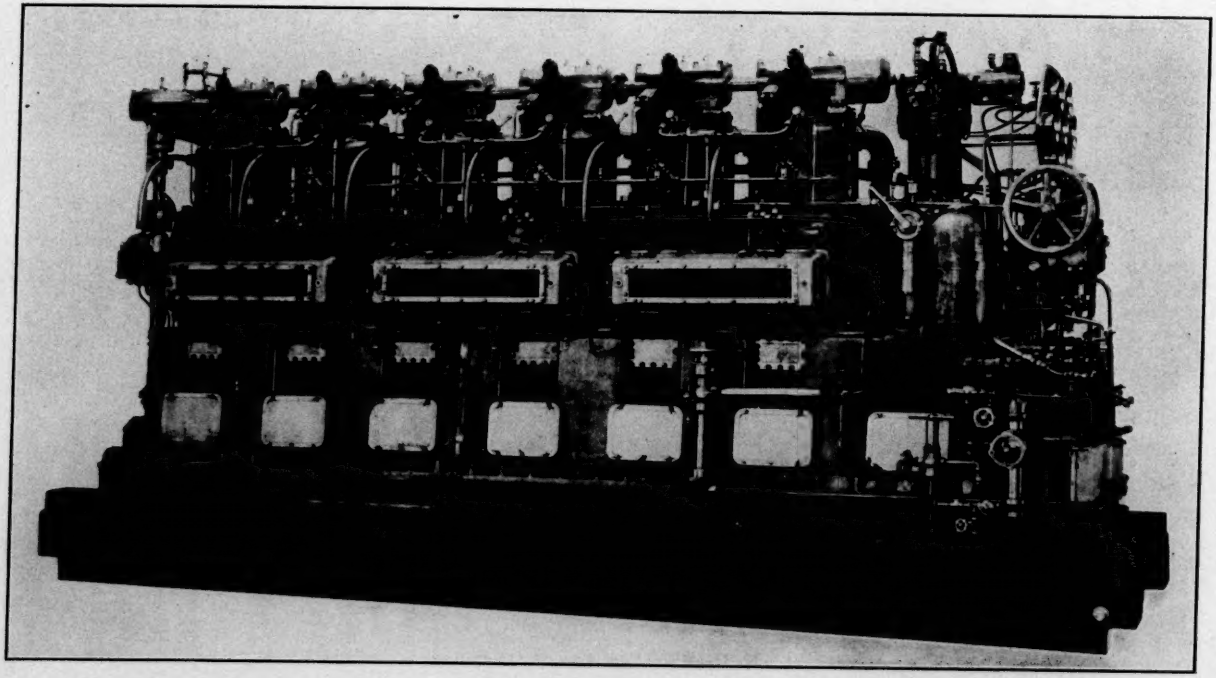


FIGURE 13.

The crank case housing was of vanadium bronze made also in two sections and bolted together. To prevent the lubricating oil from splashing on the scavenging cylinder surface, splash plates were fitted both at the bottom of the bedplate as well as around the crank and connecting rod.

The scavenging cylinder was a plain cast iron shell with flanges for suction and discharge valves which were of the automatic flat disc type. The space between the scavenging cylinder barrel and crank case housing was used as a scavenging receiver.

Suction valves were inboard and a suction muffler composed of strips of tin set on edge was originally fitted to deaden the suction noise. Discharge valves were outboard and a small aluminum jumper pipe connected them with the receiver. On the backs of these connections were fitted relief plates held by a small spring, these relief valves were set to blow when pressure inside exceeded 10 pounds per square inch.

To the top of the scavenging cylinder was bolted the cast iron working cylinder of the closed type, *i.e.*, with the cylinder head cast integral with the cylinder. A row of exhaust ports controlled by the working piston were at the lower end of the cylinder and in the top were mounted one scavenging valve, one fuel valve, one air starting valve and one relief valve. The exhaust ports led to a common chamber, surrounded by water jacket, which made up the flange for bolting to the scavenging cylinder. The holding down studs passed through steel tubes fitted through the water space. Scavenging air was brought to the scavenging valve by means of a pipe connecting it to the scavenging air receiver.

Working and scavenger pistons were one casting with the latter holding the wrist pin and serving as a crosshead. The top of the working piston consisted of a separate head carrying the piston rings, fastened to the main piston by means of studs, the nuts of which were accessible from the outside. This piston top was oil cooled, the oil being pumped through the hollow connecting rod into the wrist pin and from there

through a drilled hole in the piston wall to the piston top, discharging through another hole on the opposite side of the piston and by means of an attached pipe into a larger stationary pipe, secured to the splash plate, and leading into the bottom of the crankpit. Scraper ring was fitted into the bottom of the scavenger piston to reduce the amount of lubricating oil getting into the scavenging cylinder and from there into the scavenging air receiver. Drain valves were provided in the bottom of this receiver for the blowing out of any accumulated surplus of lubricating oil.

A two-stage air compressor was fitted with automatic disc type valves and took its suction out of the scavenging air receiver with a throttling valve controlling the amount of air sucked in. The discharges of both stages were passed through air coolers and provision was made for draining condensed water and lubricating oil.

Forward of the air compressor were the pumps—two for circulating water, one for lubricating oil and one for fuel. All the pumps were of the single acting reciprocating type driven from one crosshead, which in turn was driven from the crankshaft by a crank and spur gears at a one to two and a quarter reduction. Soft packing, held by glands, was used to pack the water and lubricating oil plungers. High speed automatic flat seat valves were used.

The fuel measuring pump, with one plunger for each cylinder, was driven by an eccentric on the main crankshaft. The amount of oil pumped to the fuel valve of each cylinder was regulated by the closing of the suction valves. An ingenious mechanism was used to accomplish this and those interested may find it described elsewhere.*

Starting and reversing were done by compressed air, the entire mechanism being operated by turning one handwheel. A rather complicated sequence of events followed but this was all done pneumatically and took a very short time.

* See JOURNAL AM. SOC. NAV. ENG., 1913, p. 447.

The operating experiences with these engines differ quite materially. Some had no particular trouble except such as might be expected from any piece of machinery running at high speed, while others had so much trouble that the engines had to be taken out and the boats re-engined. This type of engine was abandoned also by the builders and most likely for similar reasons: (*a*) It was impossible to prevent the oil vapor in the crankcase from getting into the scavenging cylinder, and the lubricating oil from splashing on the walls of the large diameter scavenging pump with the result, that the hot mixture of oil vapor and air passing through the discharge valves, would frequently explode and break the relief valve plates, which did not act fast enough to relieve the pressure. No fatalities, so far as I am able to learn resulted from this in the United States submarines, however, such casualty would cause loss of scavenging pressure and stoppage of the engine.

(*b*) There were no stand-by pumps for circulating water and lubricating oil provided until 2 years after commissioning. This meant that the attached pumps had to be kept in very good condition, otherwise they would leak and the construction of the housing around them did not prevent the salt water from getting into the lubricating oil in the bedplate and finding its way to the cooled piston top. There the salt would be deposited on the hot walls and interfere with cooling, resulting in cracked piston tops. To replace the piston it was necessary to remove camshaft, exhaust header, break all pipe connections, lift off the working cylinder, and in the cramped quarters inside of a submarine. Attempts to correct leakage around the water pump plunger by tightening the packing gland often resulted in seizure of the gland and breakage of the driving shaft.

(*c*) Taking air for the injection air compressor out of the scavenging air receiver and two-stage compression would result in frequent explosions in the air coolers—both these practices have since been abandoned.

(*d*) Pneumatic air starting and reversing gear is very attractive in theory and when handled by experts, but installed in submarines it would soon leak and could not be relied upon in emergencies,—when reversing is especially needed. The operating personnel soon learned to rely on motors for reversing.

While these experiences were being accumulated the war was going on in Europe and another class of submarines had to be provided with engines. So the same type and size of engines with some modifications were built for the "L" Class. The most important modifications were (*a*) bevel gears instead of spiral gears for driving camshaft; (*b*) two scavenging valves instead of one in the cylinder top; (*c*) holding down studs were not passed through the water jacket; (*d*) rotary pumps driven by spur gears from the forward end of the crankshaft for circulating water and lubricating oil.

These boats did good work during the war, however, their engines inherited the essential disadvantages of the type mentioned above, besides; (*a*) the oil coolers proved to be too small and the cooling coils would deteriorate and salt up oil; (*b*) bedplate drains had to be enlarged to keep the rods from splashing the oil in crankpits; (*c*) the rotary circulating water pumps wore their bushings and later their housings, diminishing their output and their packing glands were inaccessible, permitting no adjustment with the engine moving with the same inevitable result—salt in lubricating oil; (*d*) the exhaust valves of the boats would not stay tight and after diving salt water would get into cylinders and past the rings to the crankcase, with an occasional breakage of a cylinder casting.

The next development of this Nelseco,—M.A.N. two-cycle, were the engines of the *M-1*. Substantial deviations from the original construction were made; (*a*) Scavenging pump was completely divorced from the working cylinder and two double acting cylinders arranged side by side on the forward end of the engine furnished the scavenging air to a separate scavenging air receiver; (*b*) separate cylinder heads were fitted to

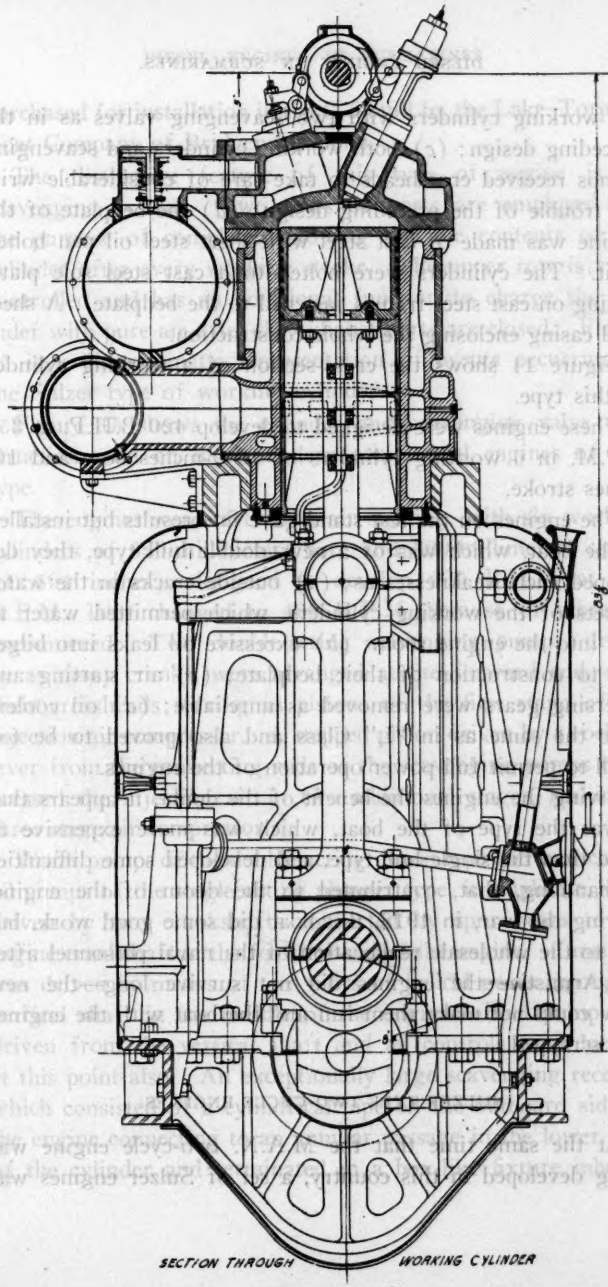


FIGURE 14.

the working cylinders with two scavenging valves as in the preceding design; (*c*) both working cylinders and scavenging pumps received crossheads to take care of considerable wrist pin trouble of the preceding design; (*d*) the bedplate of the engine was made of cast steel with sheet steel oil pan bolted to it. The cylinders were bolted to a cast steel sole plate, resting on cast steel frames fastened to the bedplate. A sheet steel casing enclosing the whole construction.

Figure 14 shows the cross-section of a working cylinder of this type.

These engines were designed to develop 420 B.H.P. at 350 R.P.M. in 6 working cylinders of 10½-inches bore and 16-inches stroke.

The engines on the test stand gave fine results but installed in the boat, which was of a new, double hull type, they developed such weaknesses as; (*a*) outside cracks in the water jackets of the working cylinders, which permitted water to leak into the engine room; (*b*) excessive oil leaks into bilges due to construction of their bedplate; (*c*) air starting and reversing gears were removed as unreliable; (*d*) oil coolers were the same as in "L" Class and also proved to be too small to permit full power operation of the engines.

Giving the engines the benefit of the doubt, it appears that it was the type of the boat, which was more expensive to build than the single hull type, and developed some difficulties in handling, that, contributed to the doom of the engine. During the war, in 1918, this boat did some good work, but due to the wholesale reallocation of the naval personnel after the Armistice the engines did not survive long—the new crew could not make them run and the boat with the engines was scrapped.

SULZER TYPE TWO-CYCLE ENGINES.

At the same time that the M.A.N. two-cycle engine was being developed in this country, a set of Sulzer engines was

purchased for installation in the *G-3* built by the Lake–Torpedo Boat Company of Bridgeport, Conn.

The distinctive feature of this type of engine is its scavenging system. Two rows of ports are employed for the purpose of completely blowing out the contents of the cylinder after every working stroke. The upper row is valve controlled and has an additional function to charge the cylinder with pure air after the exhaust ports are closed. Figure 15 is a diagrammatic representation of events occurring in the Sulzer type of working cylinder.

Figure 15 shows the cylinder and scavenging valve construction used on more modern commercial engines of this type.

The engines were reversible, air starting, with six working cylinders of 320-millimeters bore and 320-millimeters stroke (12.594 inches), designed to develop 600 B.H.P. at 450 R.P.M., later changed to 420 R.P.M. because of vibration encountered. Two double-acting scavenging pumps, driven by as many cranks, were arranged at the forward end, with their crossheads serving as pistons of the first stage of the injection air compressor. A stepped piston driven by a rocking lever from the connecting rods of the scavenging pumps compressed the injection air in two additional stages, making a three stage air compressor.

Water pumps of the double acting piston type and lubricating oil pumps of the single acting plunger type, in duplicate, were driven by two crankshafts connected by spur gears to the engine crankshaft at the forward end of the engine. Camshaft was driven by means of two pairs of helical gears through a vertical shaft in the middle of the engine. Fuel pump was driven from the vertical shaft and all controls were located at this point also. An exceptionally large scavenging receiver which consisted of a cylindrical pipe on the outboard side of the engine connecting to an annular passage in the lower part of the cylinder and terminated in a box-like fixture inboard

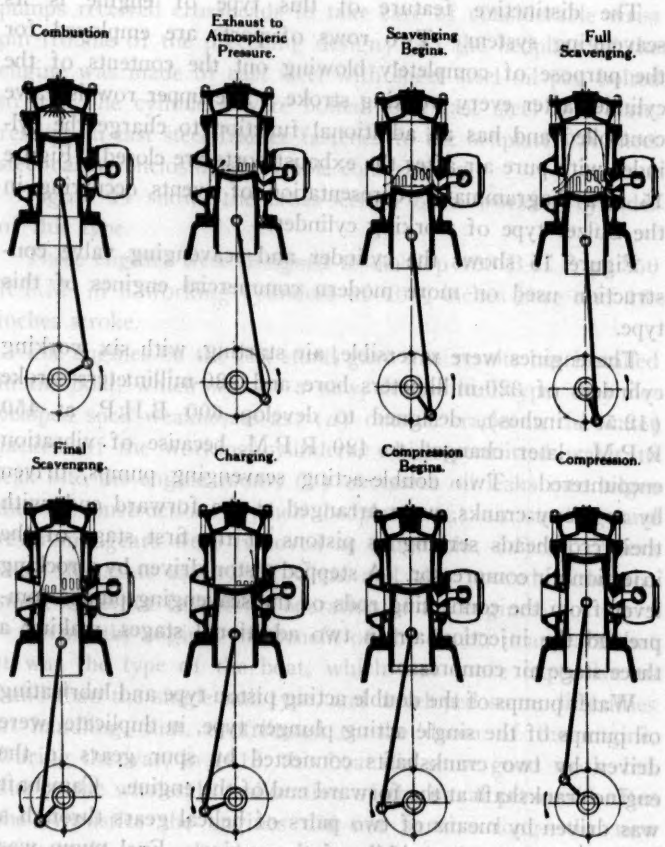

Combustion
Exhaust to Atmospheric Pressure.
Scavenging Begins.
Full Scavenging.
Final Scavenging.
Charging.
Compression Begins.
Compression.

FIGURE 15.

around the scavenging port. A double-beat poppet valve operated from the camshaft controlled the upper ports.

Bedplate was of cast bronze with lower shells of main bearings arranged for oil cooling. All cylinder heads were bolted together in one stiff unit and tie rods with shoulders served to keep the proper distance between cylinder heads and bedplate and to take the tension stresses due to the pressure in the cylinders. Cylinders were held to the heads by means of studs. The space between lower ends of cylinders and bedplate was enclosed by sheet steel doors to make an oil tight casing. Cross-rods served to give the structure lateral stiffness. A fuel valve and an air starting valve were fitted into the cylinder head, both mechanically driven from the camshaft. The well known Sulzer system of reversing was employed with two rollers mounted on two pins in a forked push rod, each of them being capable of being brought into contact with their respective cams, keyed side by side on the camshaft.

A naval inspector was sent to the works of Sulzer Brothers in Switzerland to observe the building of these engines and reported favorably on their test performance. When installed in the boat, the engines did not do so well. Apparently the engines were too large for the hull and very inaccessibly installed. Their compactness and light weight was against them too. The rocking lever drive of the air compressors soon gave trouble due to poor lubrication. Water and oil pump drive proved to be too light to stand up in service. The location of fuel pumps at the feet of the operator gave rise to numerous troubles. The installation in the boat interfered with the proper drain of the lubricating oil from the bedplate to the sump. Water and oil pumps and oil coolers were not large enough for full power operation. But worst of all was the crowded condition of the engine room, which did not permit reasonable care for weak points of the installation, besides lack of experienced operating personnel. The boat was a failure.

BUSCH-SULZER TWO-CYCLE.

The first American development of this type of engine was a 6-cylinder engine designed to develop 600 B.H.P. at 375 R.P.M. in working cylinders of 12.5 inches bore by 14.5 inches stroke, followed immediately by a smaller engine of the same type developing 300 B.H.P. at 400 R.P.M. with cylinder dimensions—9.5 inches bore by 12 inches stroke. The first was installed in "L" Class, the second in "N" Class of boats designed by the Lake Torpedo Boat Company. Whereas, the larger engine was air starting and reversing, the smaller size was neither air starting nor reversing.

Figure 16 shows the 600 B.H.P. engine as it was on the test stand at the builders works. Considerable modifications in the design were introduced, based partly on the experiences with the engine in the *G-3* and partly on the operating results with the stationary engines of American manufacture. The following are the more important ones; (*a*) working cylinder of the closed type with an integral cylinder head was chosen, accordingly the tie rods were made shorter, reaching only to the rectangular entablature around the exhaust and scavenging ports, where all the cylinders were tied together; (*b*) the scavenging pump consisted of two double acting cylinders arranged in tandem and driven from one crank immediately forward of the working cylinders with the lower piston of the differential type serving also as a crosshead. One piston valve driven by an eccentric from the crankshaft controlled suction and discharge of all four cylinder ends; (*c*) the scavenging pump discharged into one common box-shaped scavenging air receiver on the inboard side of the engine. The double beat poppet valve for the control of upper ports was retained; (*d*) a standard type of three-stage air compressor with differential piston was driven by the forward crank of the main shaft with straight tube air coolers arranged on either side of it; (*e*) location and type of circulating water and lubricating oil pumps was retained except that all

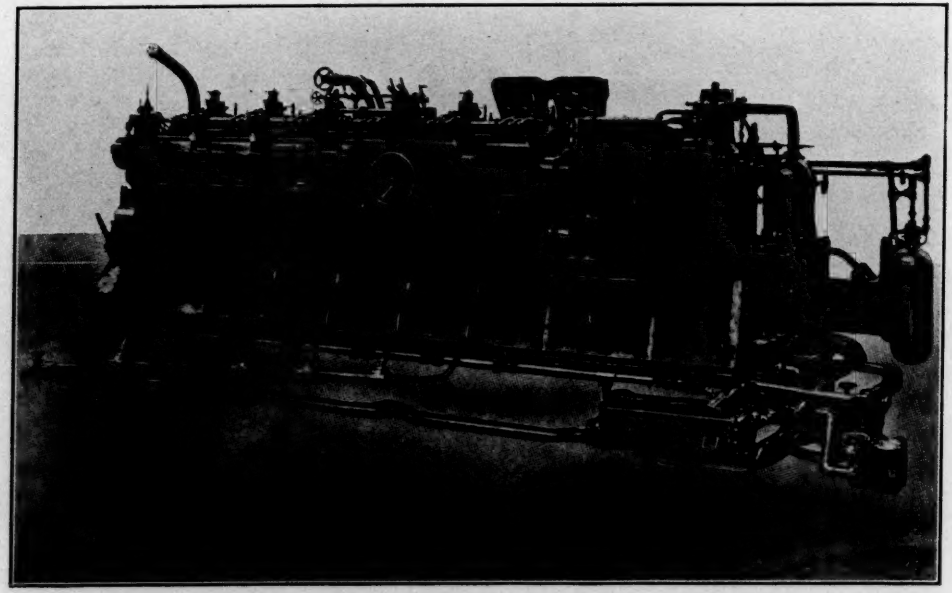


FIGURE 16.

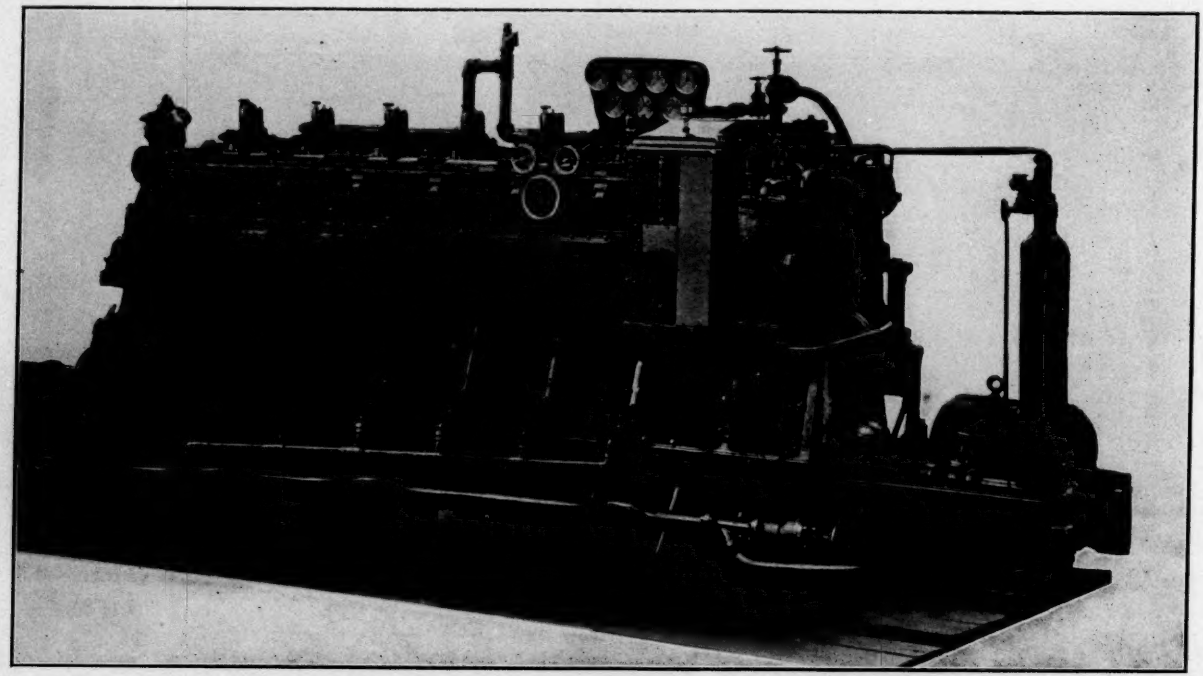


FIGURE 17.

of the pumps were connected to one crosshead and driven by a jackshaft geared to the main shaft; (*f*) fuel measuring pump was driven by the vertical shaft located on the aft end of engine with the overspeed governor mounted on top of this vertical shaft.

Figure 17 shows the 300 B.H.P. engine which had a gear driven rotary type lubricating oil pump attached to the forward end of the engine. A motor driven centrifugal pump supplied circulating water. These small engines had very little trouble except the differential piston of the scavenging pump which required watching and good lubrication. At any rate they outlived their hulls and were reinstalled in the "L" Class of boats instead of Nelseco—M.A.N. two-cycle engines.

The larger 600 B.H.P. engines did not escape, however, considerable trouble caused mostly by weak pump drive at the forward end and inefficiency of oil coolers which were manufactured by the engine builders and were a continuous source of trouble. But even this deficiency in oil coolers did not result in any major casualties to working cylinders and pistons. Air compressor valves which were of the mushroom type would break at the stem due to poor guiding of the small diameter stem.

FOUR-CYCLE ENGINES.

The success of the 120 B.H.P. 4-cylinder, 4-cycle commercial engine developed by Nelseco was the beginning of a long period of preference among the operating personnel of the Navy for the 4-cycle engine and resulted in the building of three distinct types, with several sizes of each type, for installation in United States submarines. All of these types of engines are still in active commission and although some of the installations gave considerable trouble, these difficulties were not beyond elimination.

NELSECO FOUR-CYCLE.

A set of eight cylinder engines with two injection air compressors at the forward end which were practically duplicates

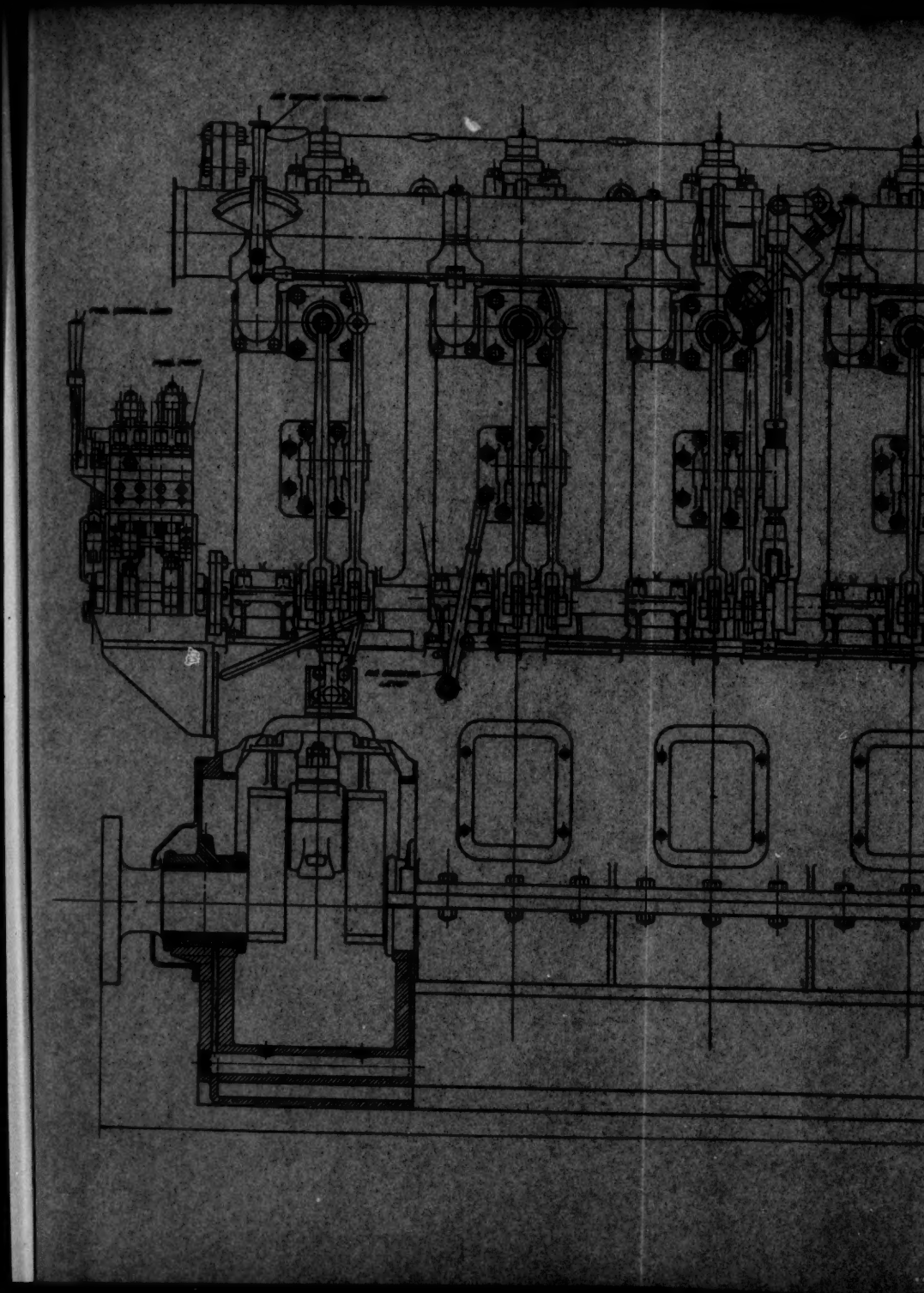
of the above referred to commercial engine, except with twice as many cylinders, were first installed in the Holland type "N" boats and later used in re-engining the "H" and "K" Classes. Similar engines of 4-cylinders replaced gasoline engines in "D" boats and a set of 6-cylinder engines replaced the original Vickers type of engine in the *E-1*.

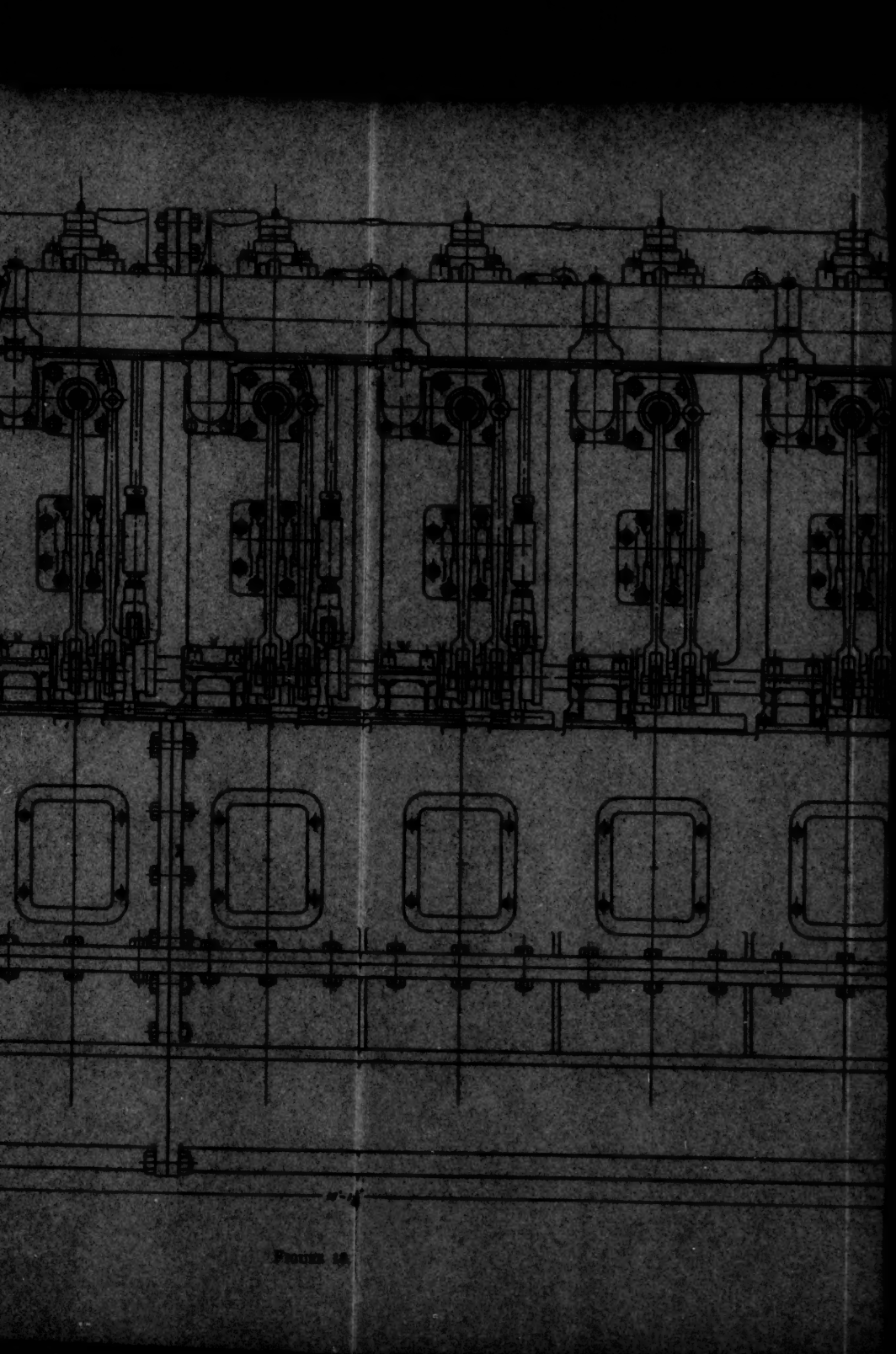
These engines were designed to develop about 30 B.H.P. per working cylinder at 350 R.P.M., these cylinders being of 9-inches bore and 12½-inches stroke.

Figure 18 shows an outside view of the eight cylinder unit and Figure 19 shows cross-sections.

The most distinctive feature of this particular type of engine is the shape of the combustion chamber—originally extensively used in horizontal gas and oil engines with vertical valves. In this case the valves are arranged horizontally and driven by two separate camshafts, one on each side of the engine. The original engine is of a rather heavy type with both bedplate and the box-shaped crank case made of cast iron. The cylinder head is cast integral with the cylinder and in order to inspect the piston the whole cylinder has to be removed. The inlet valve has a separate cage whereas the exhaust valve has only a separate seat pressed into the cylinder and the exhaust valve is withdrawn through the inlet valve cage opening. Duplicate two-stage air compressors are mounted on the forward end, the details of which are clearly shown in Figure 19. The camshafts are driven by means of spur gears through an idler—the drive being at the forward end of the engine. Fuel pump and governor are driven from the aft end of the camshafts.

These engines are non-reversible, however, the four center cylinders are equipped with mechanically operated air starting valves. Separate motor driven water and lubricating oil pumps are used with these installations. As mentioned before, these engines proved to be quite a success and are well liked by the Navy personnel. The boats are stationed at the submarine school for instruction purposes and almost every offi-





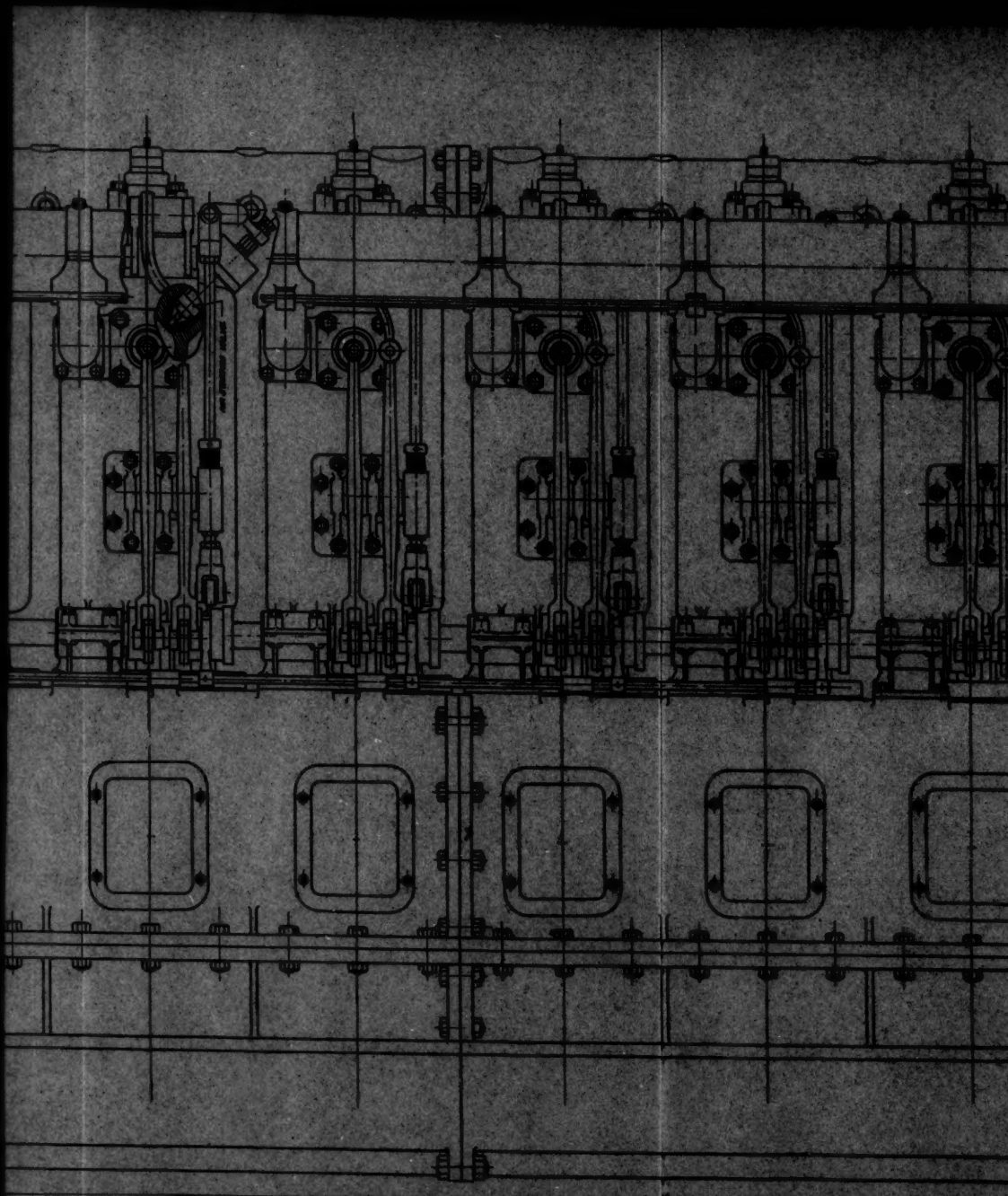
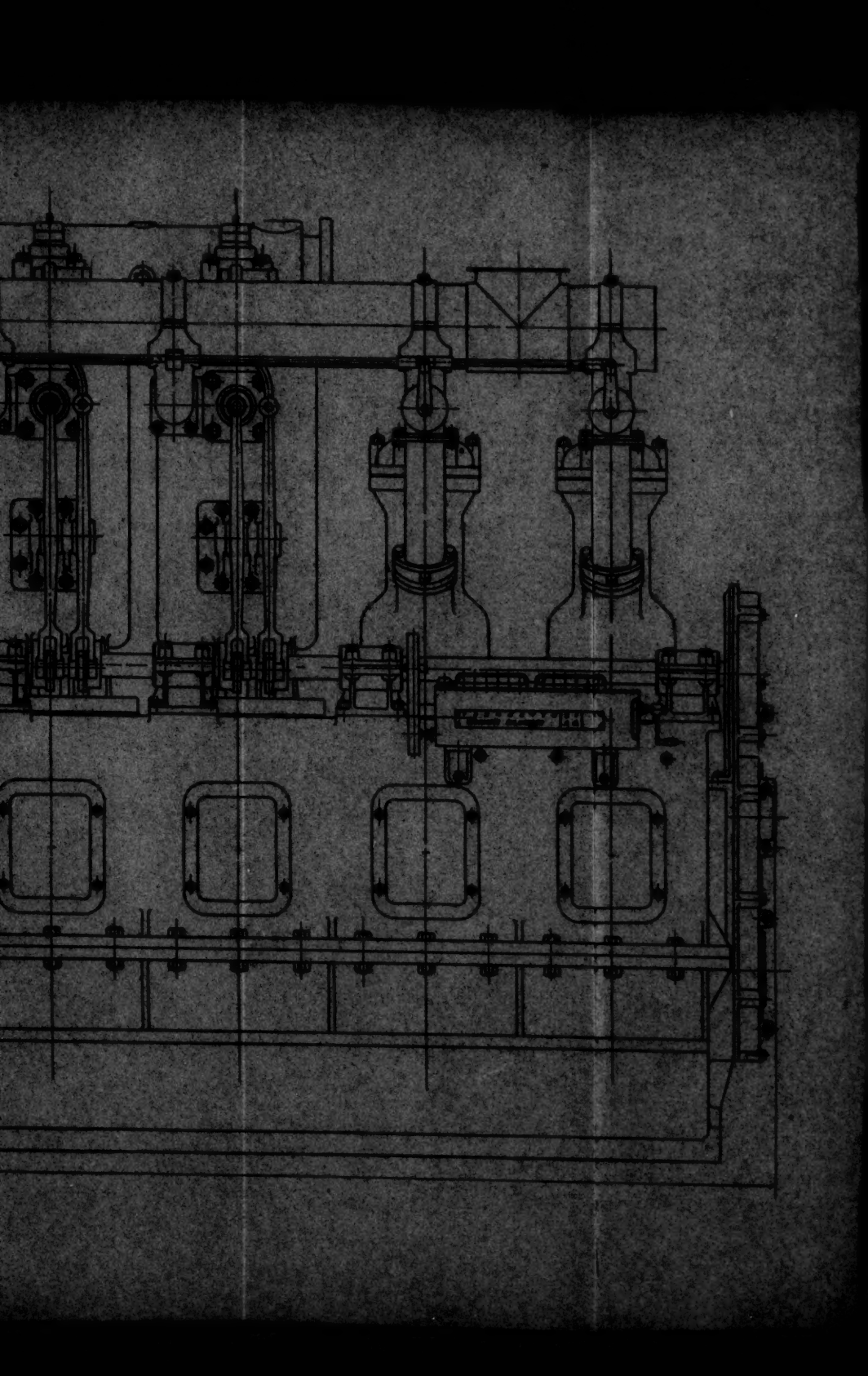
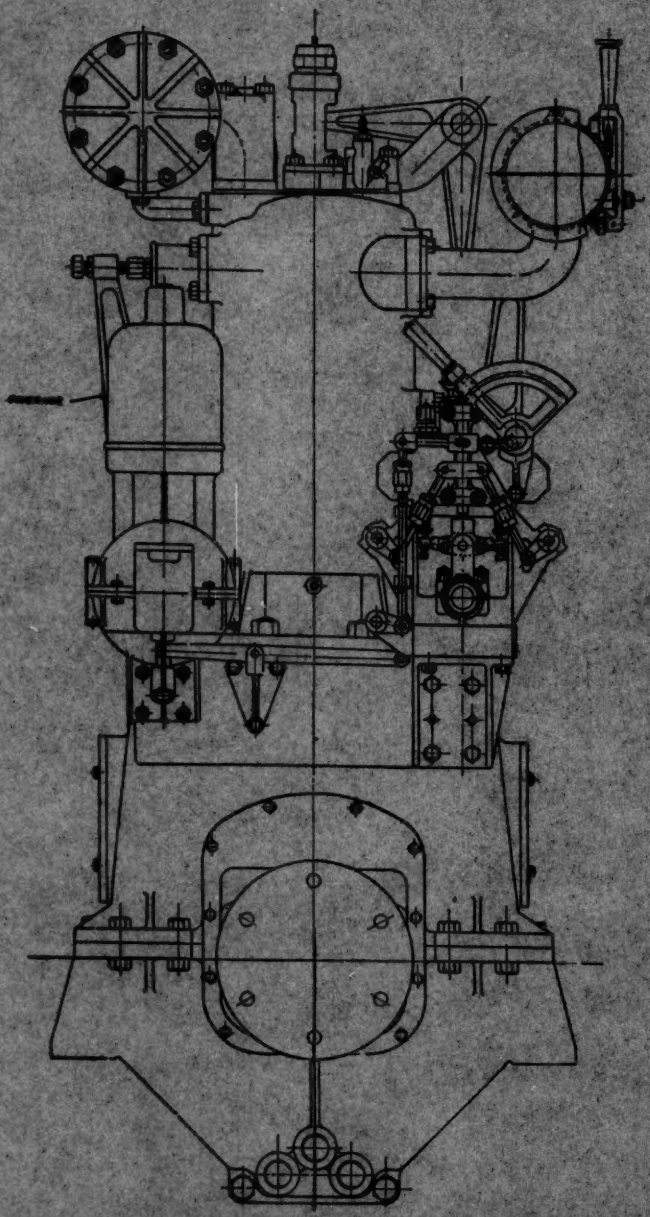
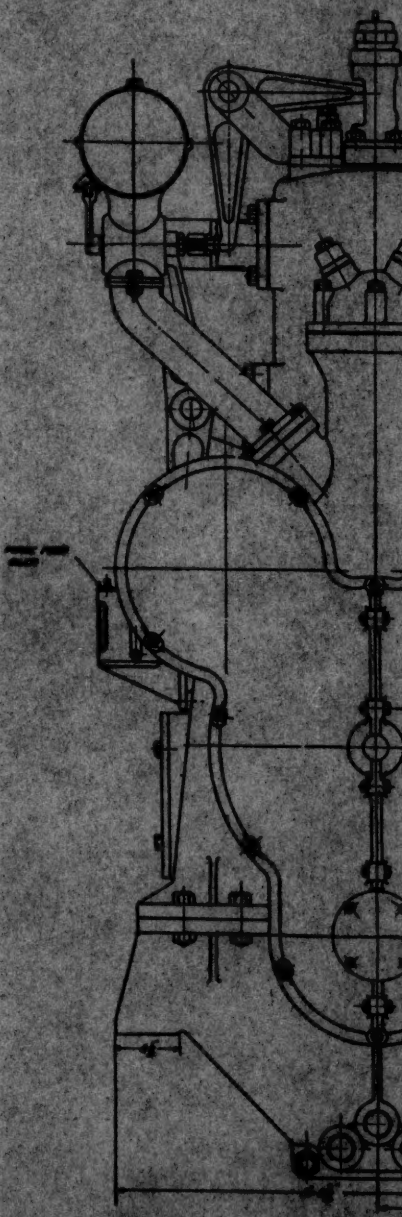


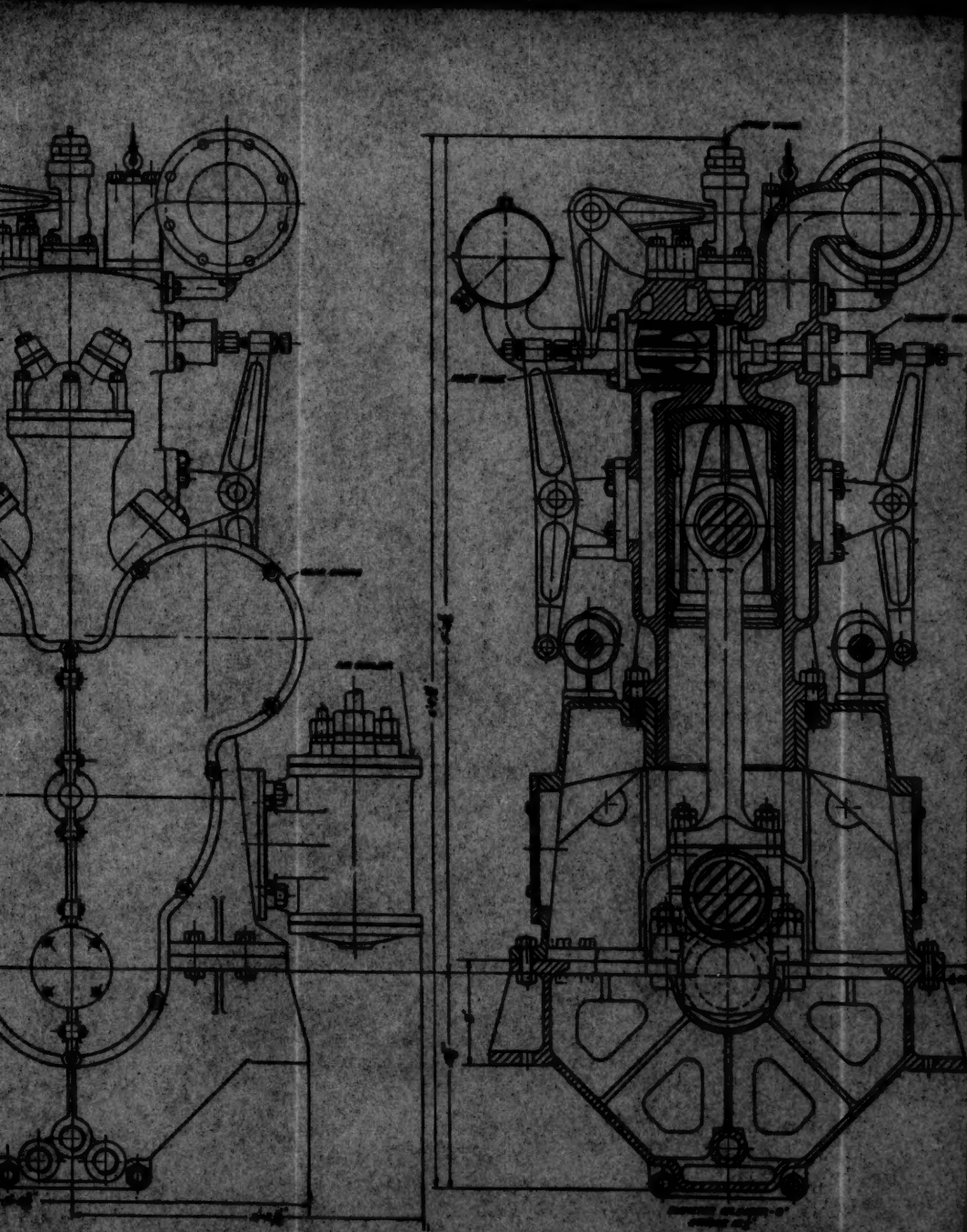
FIGURE 18





AFTER END





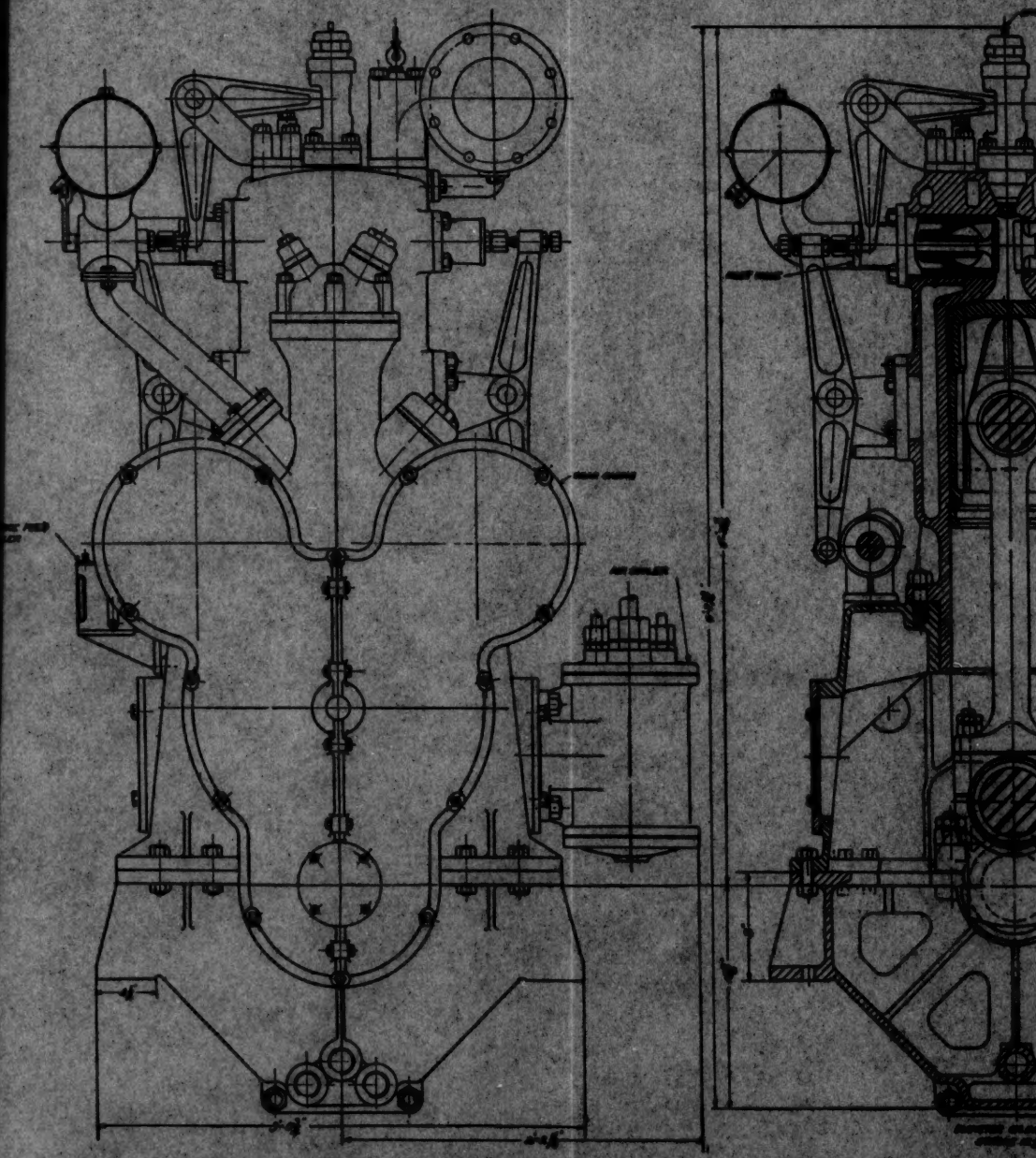
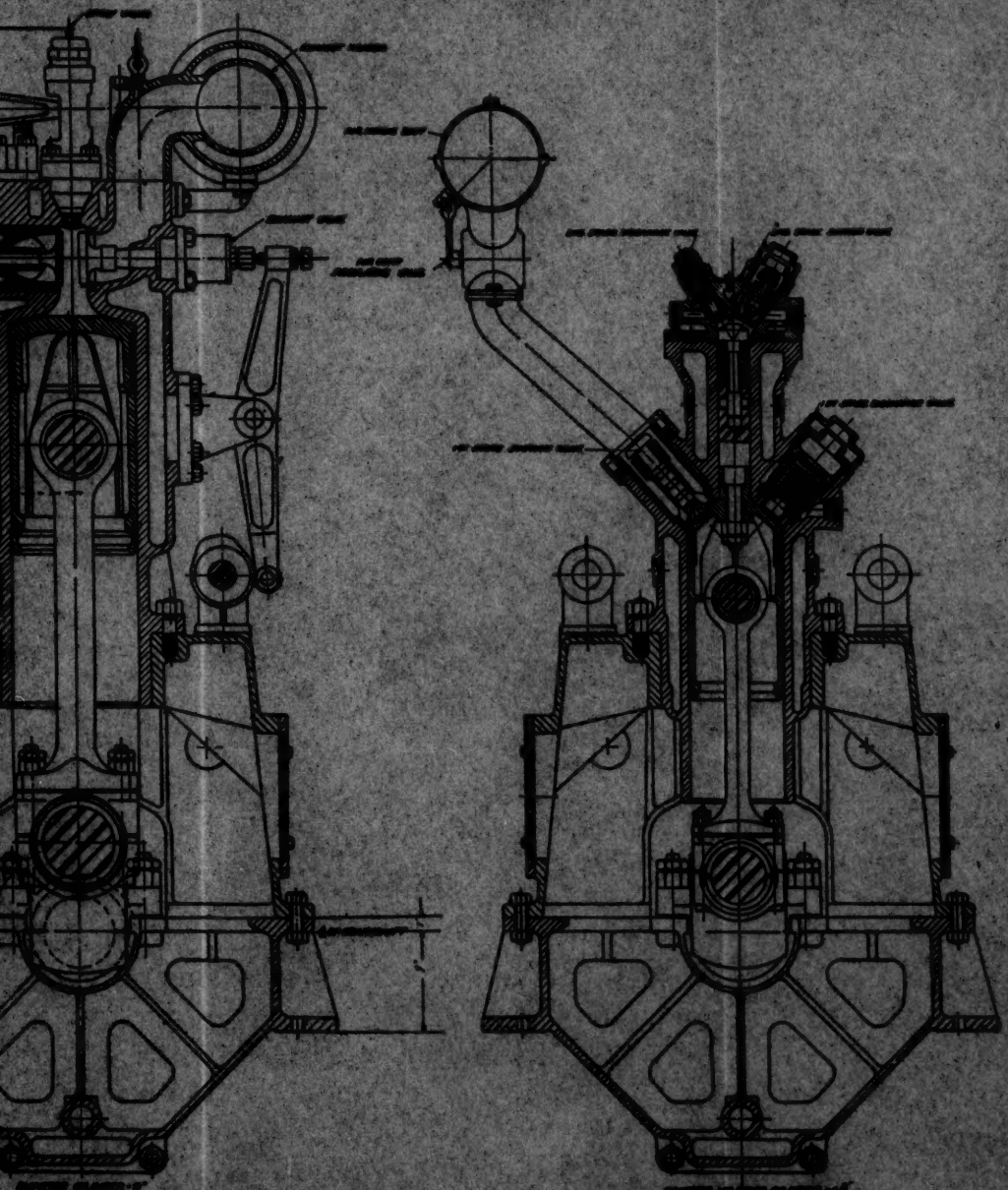
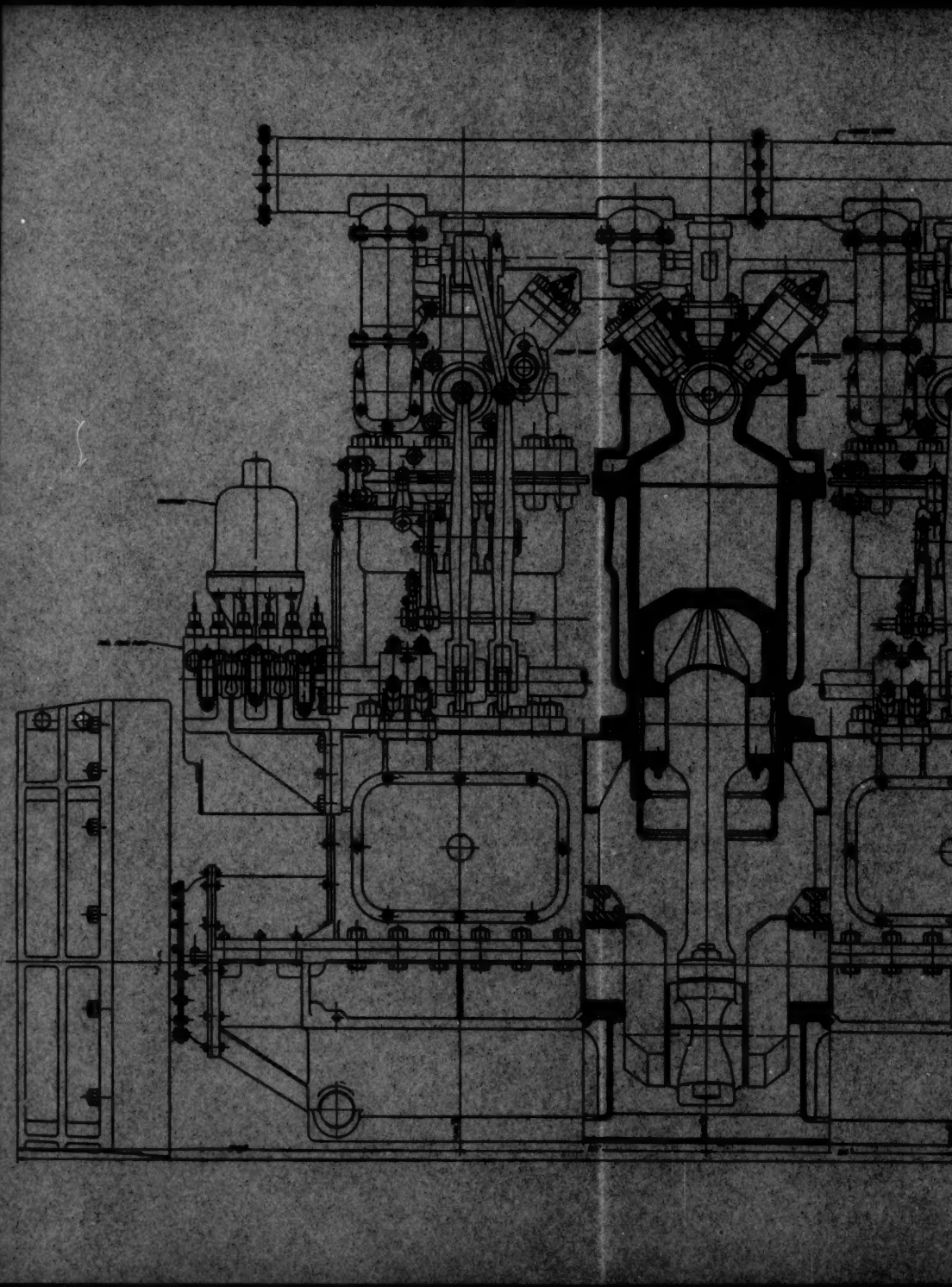
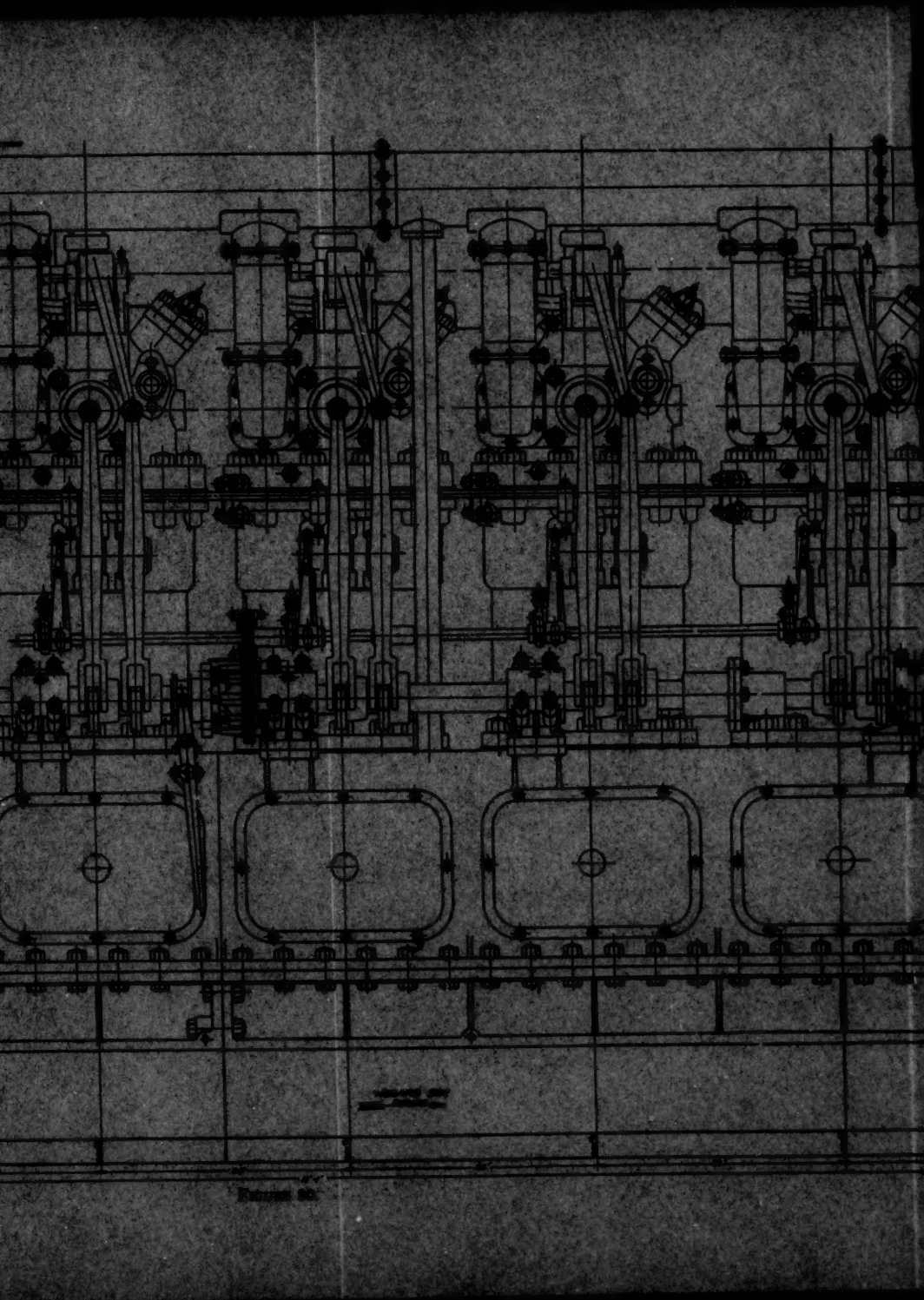


FIGURE 19.







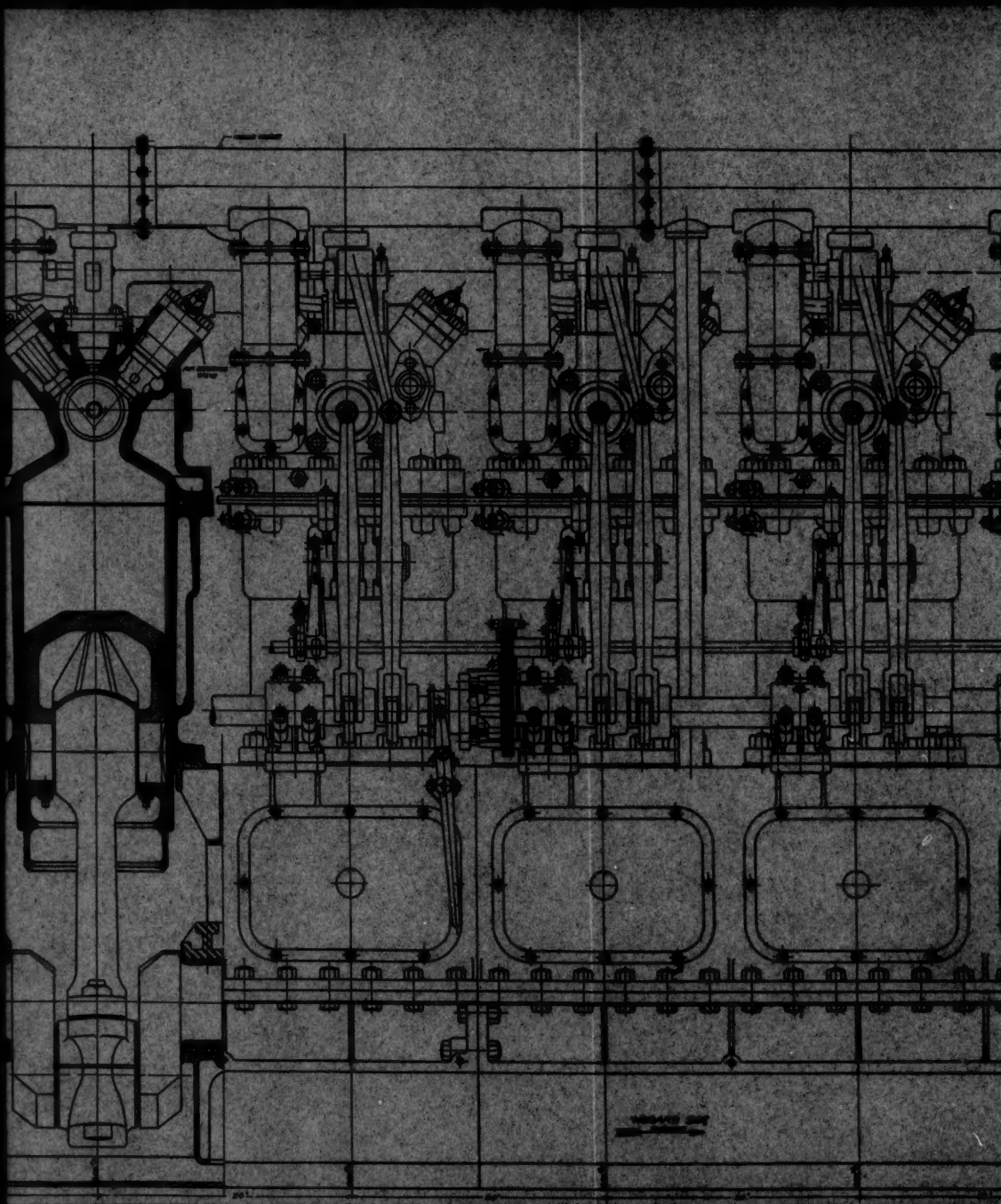
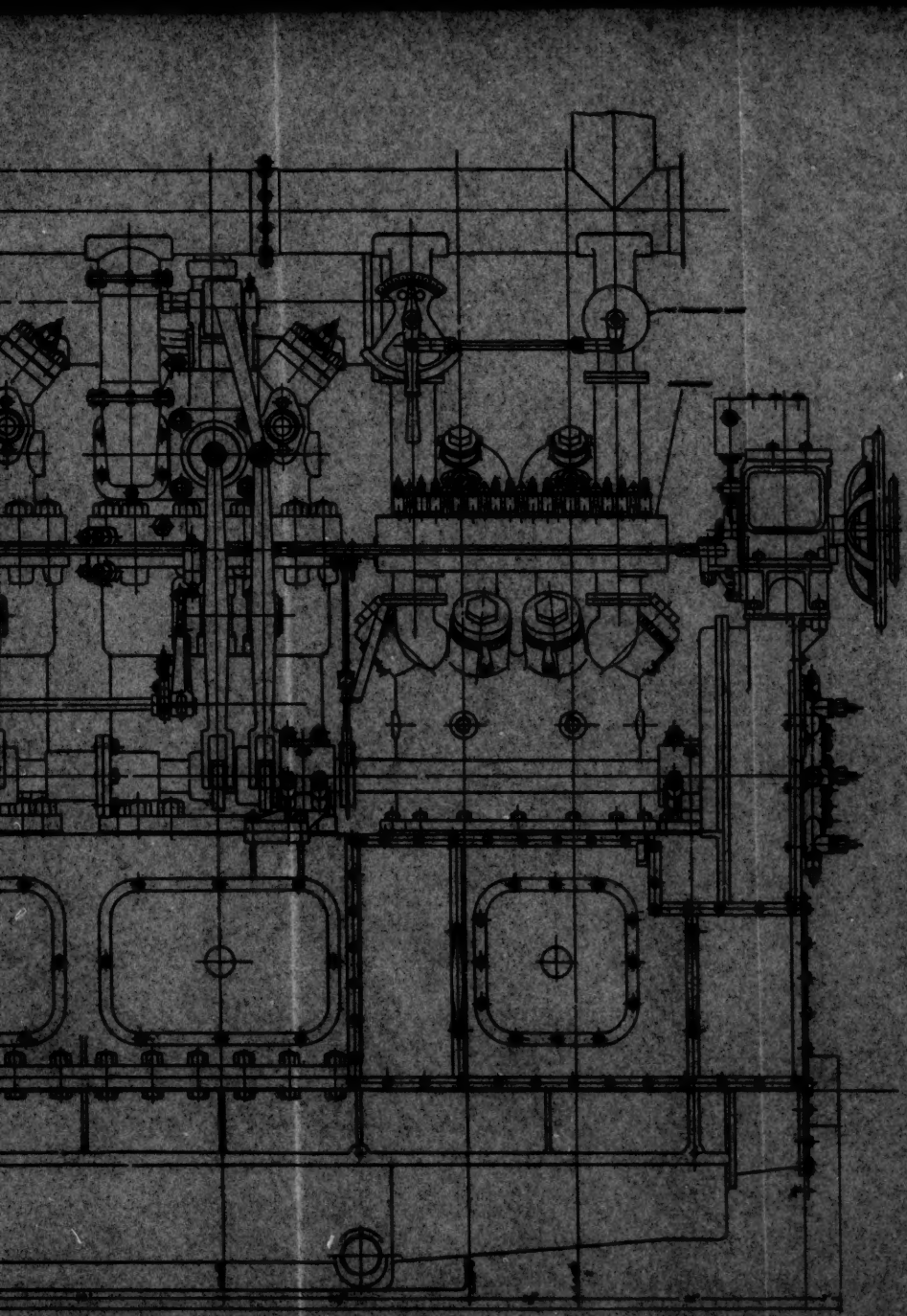
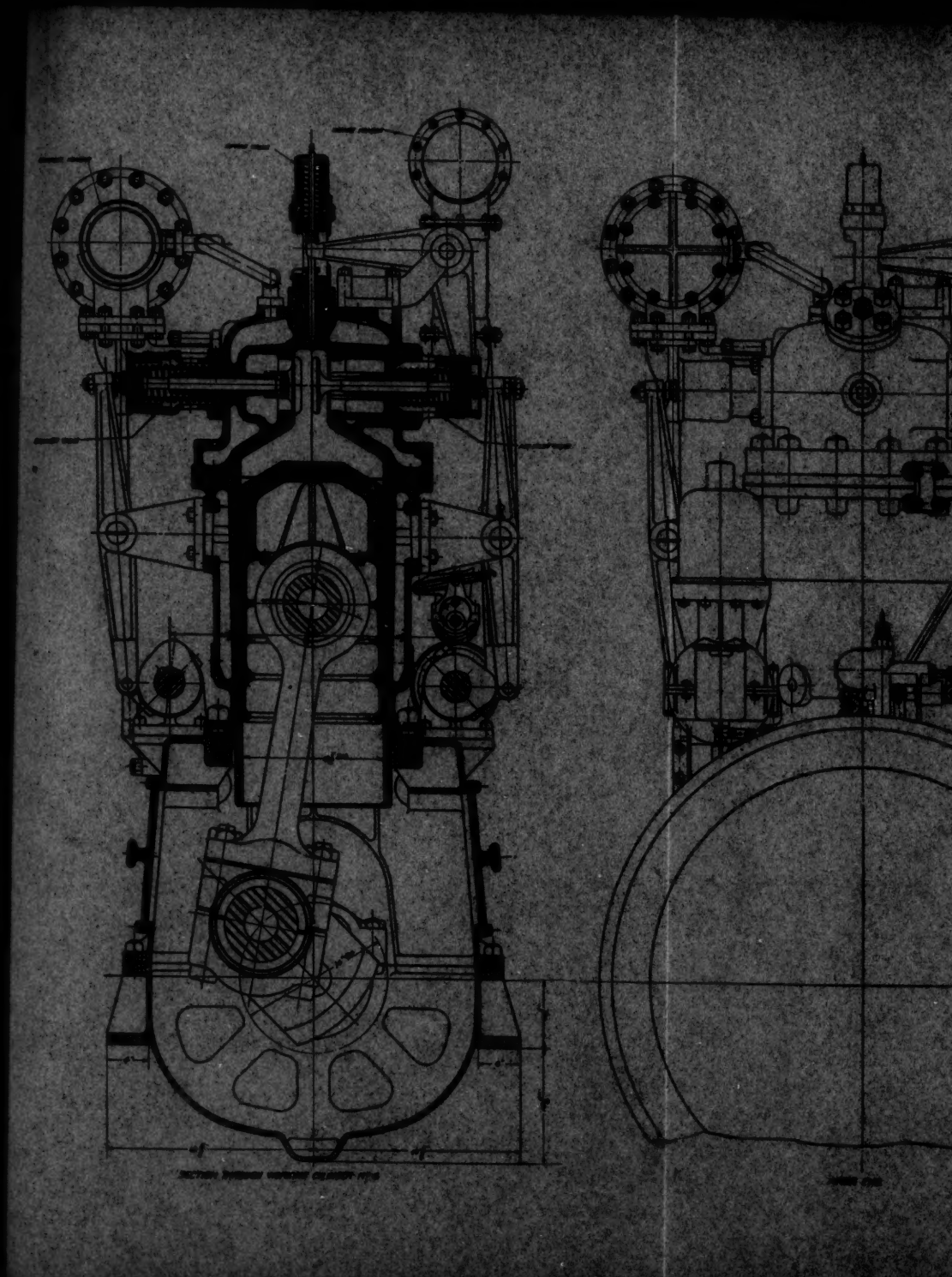
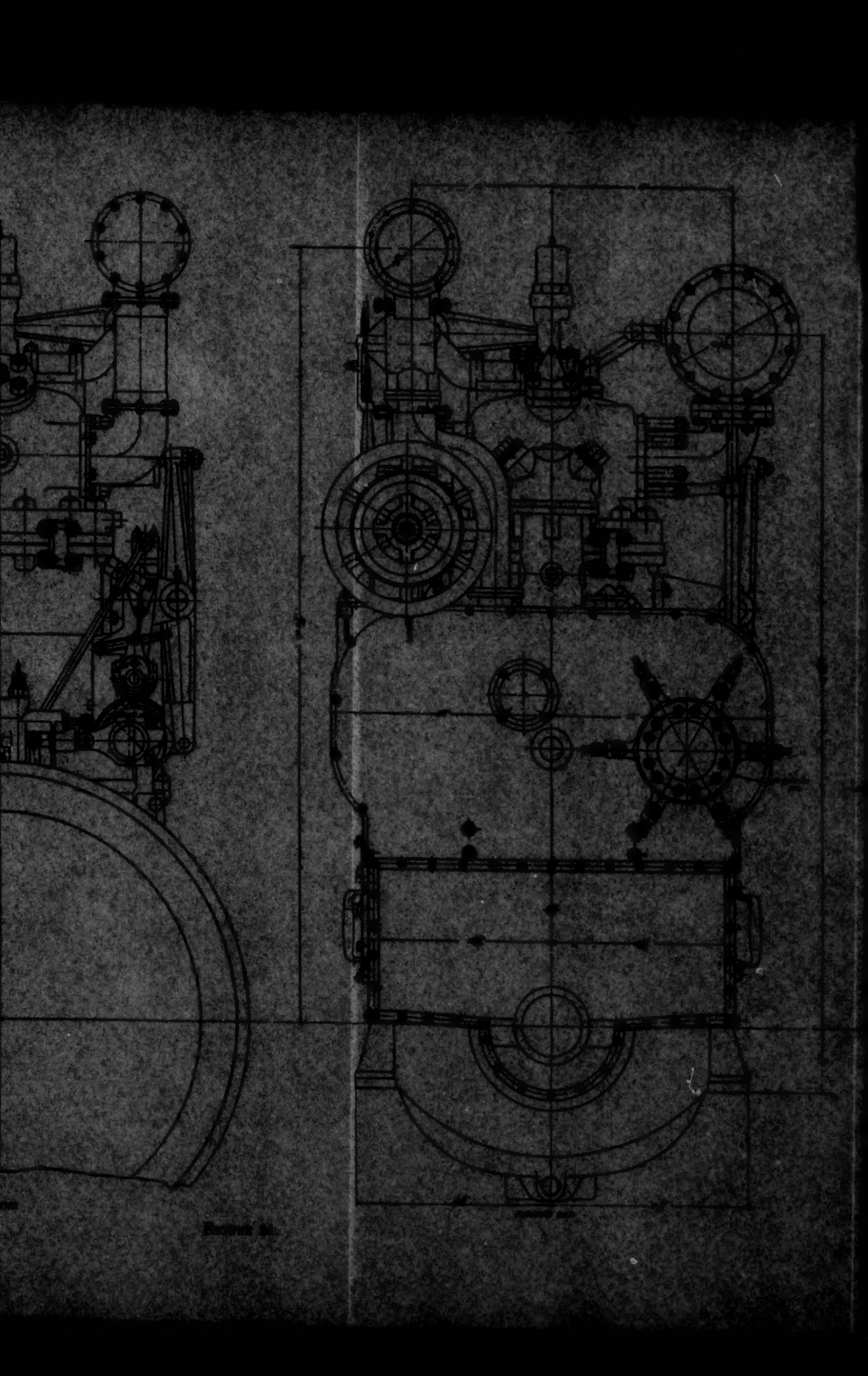


FIGURE 20.







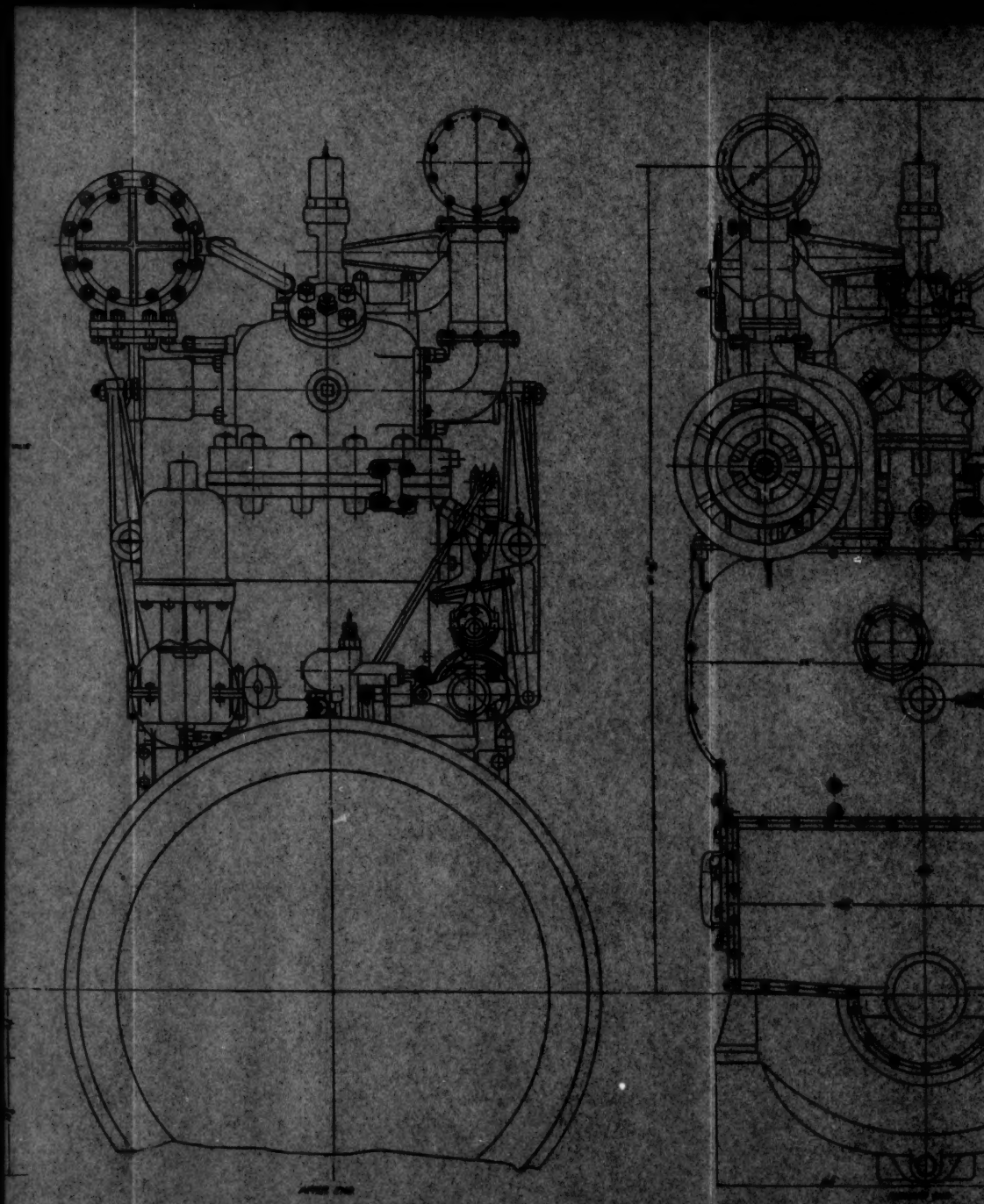
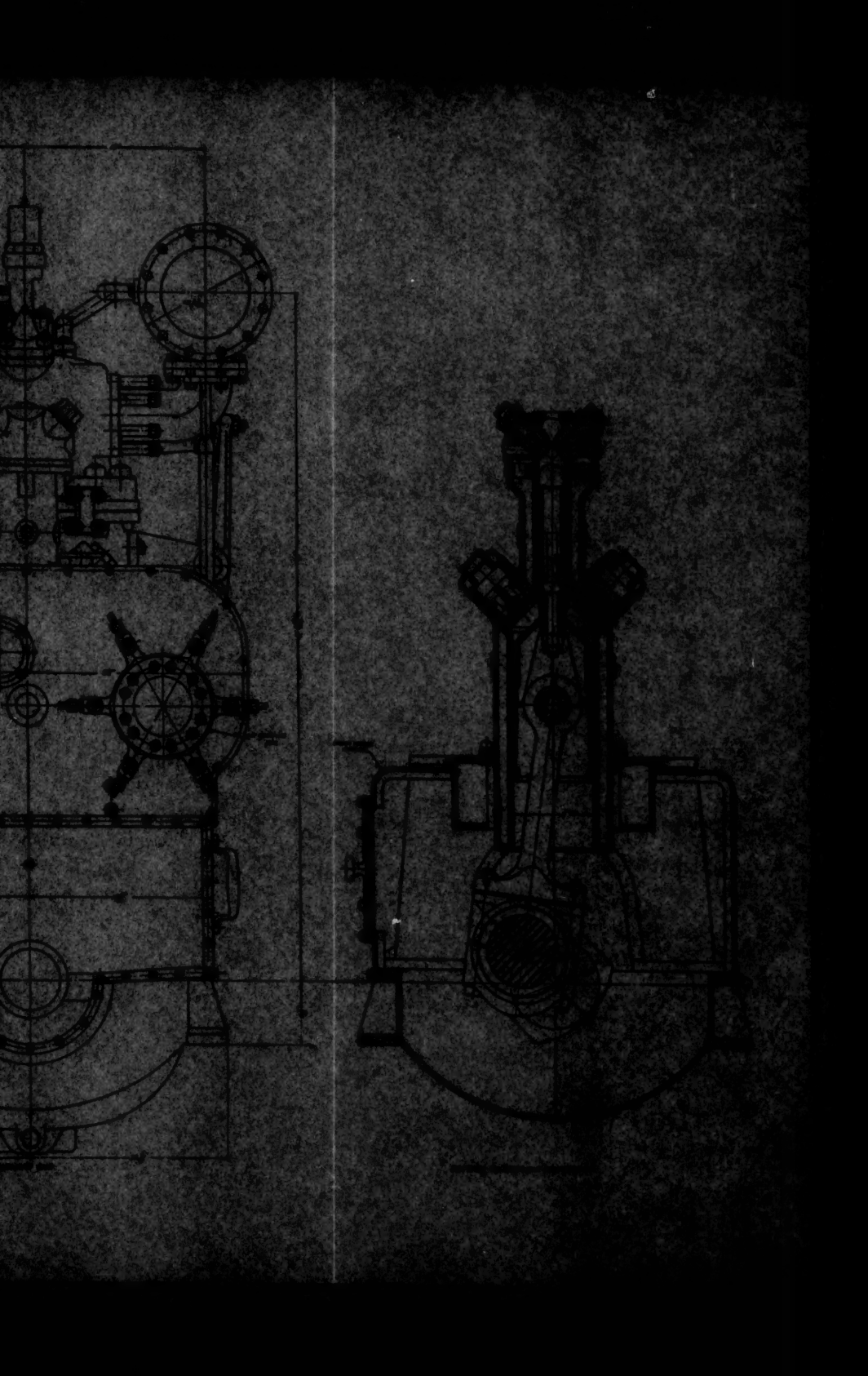


FIGURE 84.



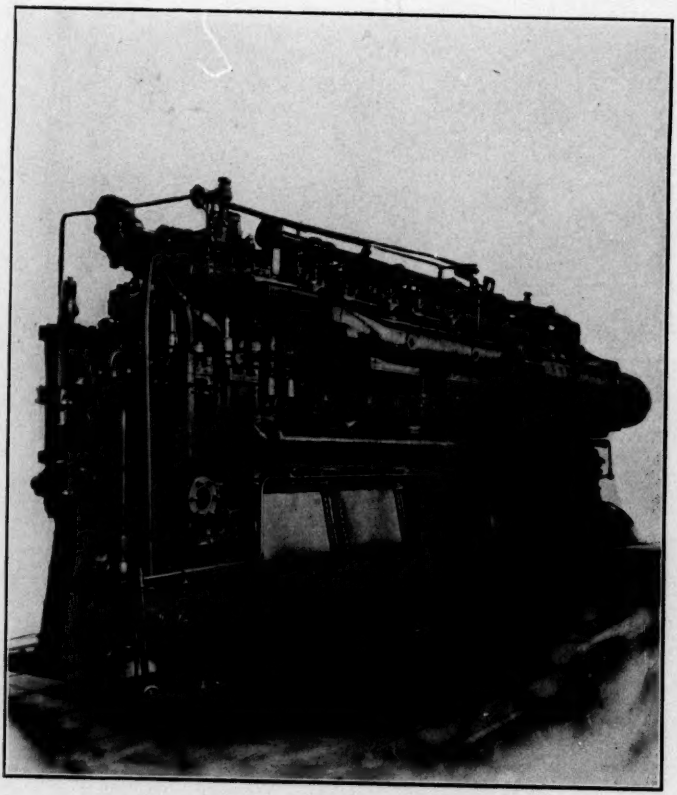


FIGURE 22.

cer and enlisted man in the submarine service today has at the beginning of his submarine career operated these engines.

Based on the experiences with the above engine, several more different sizes were designed. "O" and "R" Classes of submarines have six cylinder engines developing 440 B.H.P. at 400 R.P.M. with 13½-inches bore and 14-inches stroke. The only difference between the two is that the later engines received air starting; both are non-reversible.

Figure 20 shows an outside view and Figure 21 sections of this type of engine. Separate cylinder head was adopted, twin two-stage injection air compressors made in one block at the forward end of the engine and, to reduce weight, both bedplate and crankcase were made of bronze. On the later engines, as an afterthought, tie rods were fitted inside the crankcase to relieve the crankcase of some of the tension stresses due to the recurring cylinder explosions. Long experience, however, proved that bronze was not a suitable material for crankcases, and that the tie rods under service conditions hardly worked as they were intended to and therefore cast steel crankcases are replacing the original ones. This is the only major casualty with this type of engine except possibly those due to overloading which will be discussed later. Here too, separate motor driven pumps are used. Both engines and boats proved to be very reliable and efficient despite their continued service since 1918 and 1919 when the two classes of boats were commissioned.

BUSCH-SULZER—FOUR-CYCLE.

The Busch–Sulzer Company was prevailed upon by the Navy Department to design and build a 6-cylinder four-cycle, non-air-starting and non-reversible engine developing 500 B.H.P. at 410 R.P.M., with cylinders of 14¼-inches bore and 14½-inches stroke. Figure 22 shows the outboard side of this type of engine.

The same mechanical construction that characterizes all Busch–Sulzer engines was retained. Bottoms of working

cylinders were expanded into stiff box-shaped structures held together by fitted bolts through suitable flanges and serving to transmit explosion stresses onto the tie rods fitted between the cylinders. These tie rods passed through cast steel spacers or cylinder supports bolted to the cross girders of the bedplate and served as lateral stiffeners of the engine structure. Suitable sheet steel doors enclosed the moving parts of the engine and made up a gas and oil vapor tight crankcase. This construction permits, quick withdrawal of the piston through the crank case by removing light bolted skirts from the bottom of the working cylinder, easy access to all moving parts within the crankcase, and found great favor with the operating personnel.

Bedplate is made of bronze and the main bearings have their lower bronze shells fitted for oil cooling. The cylinder head is of the conventional type with vertical valves carried in separate cages, the exhaust valve cage being water cooled. To reduce the length of the cylinder studs, these are carried by the head and go through the flange on the cylinder with nuts set in pockets. The camshaft is mounted in a casing carried by the top face of the cylinder head and is driven by means of a pair of helical bevel gears from the vertical shaft which is driven by a pair of helical gears from the aft end of the main crankshaft. The fuel measuring pump and the cylinder lubricating oil pump are driven from the vertical shaft. A three-stage injection air compressor of the conventional differential piston type is driven by a separate crank on the forward end. On the first engines built a rotary lubricating oil pump was driven by gears from an extension on the forward end of the crankshaft.

Three separate groups of these engines were built and installed in "O," "R" and some of the "S" boats built by the Lake Company. Minor differences exist between them, but space does not permit mentioning them, except that the "O" boat compressor was greatly improved by the subsequent

design using plate valves. The exhaust from the forward three cylinders was made not to interfere with that of the aft three. This made a very difficult casting of the aft portion of the exhaust header, shown on Figure 22, and caused considerable trouble in service due to cracking. The construction was abandoned and the "S" boats received a much simpler header which also was furnished as replacements for the "R" boats. The only other outstanding objection the service has to this type of engine is the fuel measuring pump.* As mentioned already, this pump is driven by the vertical shaft which in this case runs at engine speed. This was done, obviously, in order to reduce the size of the pump parts as well as governor weights. In a four-cycle engine, fuel is injected into the cylinder every two revolutions and the pump therefore makes two deliveries of half the amount necessary for each working stroke of the cylinder. All the six plungers of the pump are divided into two groups and each group is fastened to opposite sides of a stiff rectangular frame driven by one eccentric. This arrangement makes the plungers pump 180 degrees apart whereas the cylinders fire every 120 degrees, hence it becomes inevitable that two pump deliveries will be made during the exhaust stroke of some of the cylinders. If all the fuel valves are in good condition, all is well, but should the pump deliver into a leaky valve on the exhaust stroke of that cylinder—the fuel will be blown into the cylinder where it will partially burn and increase the temperature of the exhaust gases much to the distress of the exhaust valve. It probably keeps on decomposing even in the exhaust header and comes out as black smoke from the muffler. As a result of such design of fuel pump, the engines require more care and attention on the part of the operators and forcibly advertise the lack of it, which on long cruises becomes inevitable. In all other respects the engines stand up well in the service and have made a host of friends in the Navy. A recent

* See Factors in Design of Fuel Measuring Pumps for Diesel Engines—by J. E. Canoose—JOURNAL AM. SOC. NAV. ENG., Feb. 1925.

cruise of the "S" boats equipped with these engines back from the Philippines where they had about two years of arduous duty is a tribute to this type of engine.

S–BOAT ENGINES.

While the war in Europe was still raging and even prior to the entry of the United States into the war, three different types of "S" boats were authorized with different propelling plants. While *S-1*, a Holland type of boat, received Nelseco air starting, reversible 8-cylinder four-cycle engines, designed to develop 600 B.H.P. at 380 R.P.M., with cylinder dimensions of 13.5-inches bore and 15-inches stroke. The *S-2*, a Lake type of submarine, was equipped with air starting, non-reversible, 6-cylinder, two-cycle engines, of Busch–Sulzer manufacture, designed to develop 900 B.H.P. at 350 R.P.M., with cylinders of 14-7/8-inches bore and 16-3/8-inches stroke; the third boat the *S-3*, a government designed hull, was to be propelled by air starting and reversible 8-cylinder, 4-cycle engines of Nelseco design and manufacture, the Government purchasing in addition, the right to build duplicates of them at the New York Navy Yard. These engines were designed to develop 700 B.H.P. at 350 R.P.M., with cylinders of 14.5-inches bore and 16-inches stroke.

NELSECO—8-CYLINDER, 4-CYCLE ENGINES.

As far as the details of design are concerned, both the 8-cylinder engines above mentioned were a repetition of the previously described six-cylinder one, except that they were made air starting and reversible. Figure 23 shows the elevation of one of these engines from which the details of the reversing gear can be followed. Three separate camshafts were used—one outboard for exhaust, one lower inboard for inlet, and upper inboard for fuel valves. Reversing as well as air starting was done pneumatically. By turning the smaller wheel, the engine was first stopped by stopping fuel injection and air supply; by the movement of the large wheel, air was

admitted into double acting air cylinder, the piston rod of which was extended into another double acting cylinder, filled with glycerine which acts as a dash-pot. The movement of the piston in the air cylinder was transmitted to a forked lever engaging the crossheads of collars on the three camshafts. The camshafts had helical grooves and the collars had a narrow slotted ring acting as a nut in these grooves. Longitudinal movement of these collars caused by the air acting on the piston would result in suitable angular displacement of the camshafts. With the camshafts thus ready for operation in reversed direction, the air passes through the reversing cylinder to a relay valve and the starting control valves in the star shaped control box on the forward end. The opening of the relay valve admits starting air to the starting valves of all the cylinders and further movement of the large wheel will finally locate the cylinder that is ready for starting—the starting cam will permit the control valve to admit air to the top of a plunger in the starting valve, resulting in the opening of the valve and admission of starting air. When the engine begins to turn, the operation of the small hand wheel will admit fuel and injection air. This is only a skeleton description of the system which also includes venting of cylinders and pneumatic shutting off of injection air and fuel pump to prevent any accidents.

Such an elaborate system of maneuvering soon proved itself much too delicate for submarine service conditions and too difficult to keep in order and if maneuvering gear cannot be absolutely relied upon it has no place in a submarine. Therefore, this gear was soon removed and even the air starting part of it is not used except where special efforts are made to keep it tight and functioning.

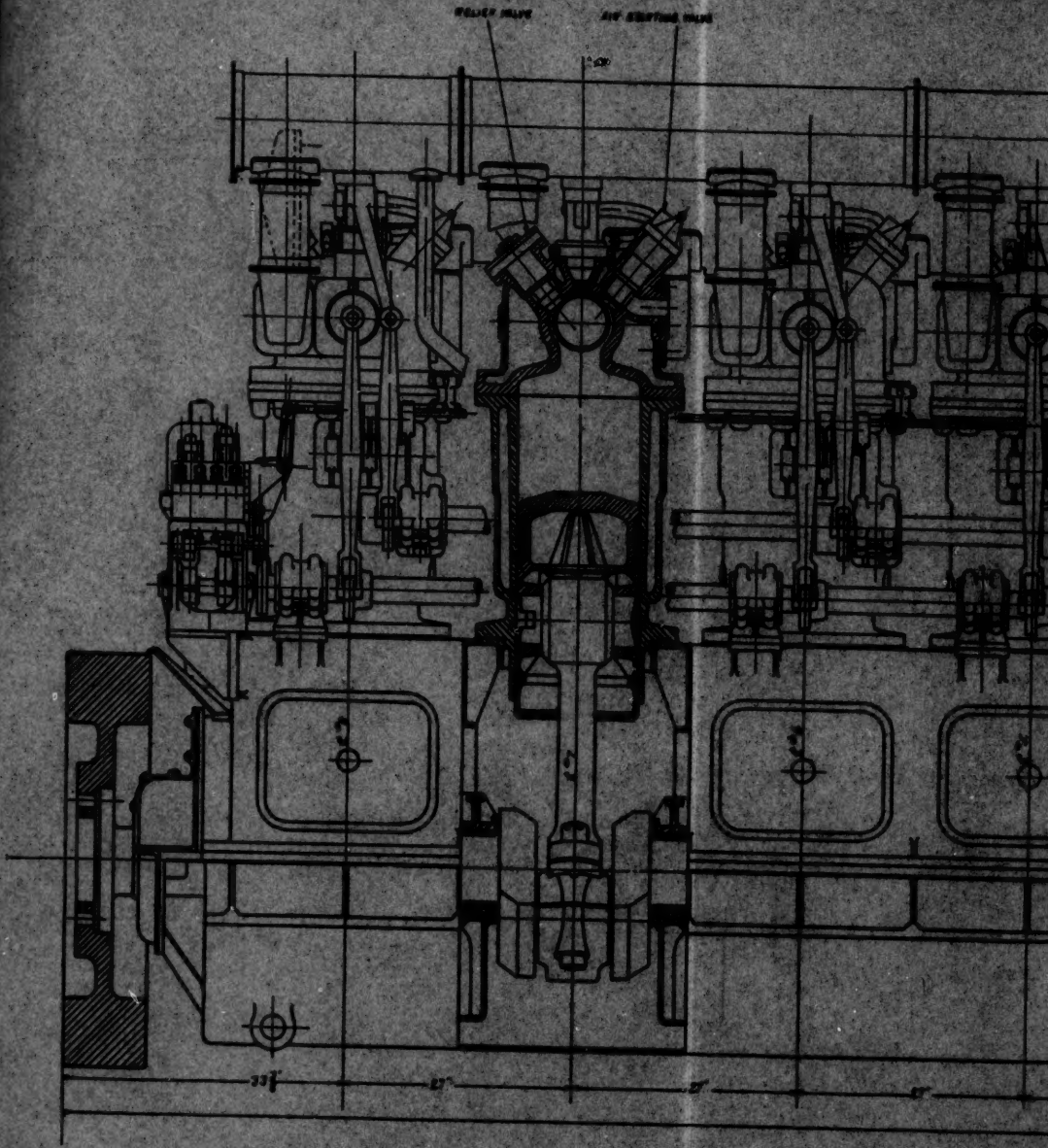
CRITICAL SPEEDS.

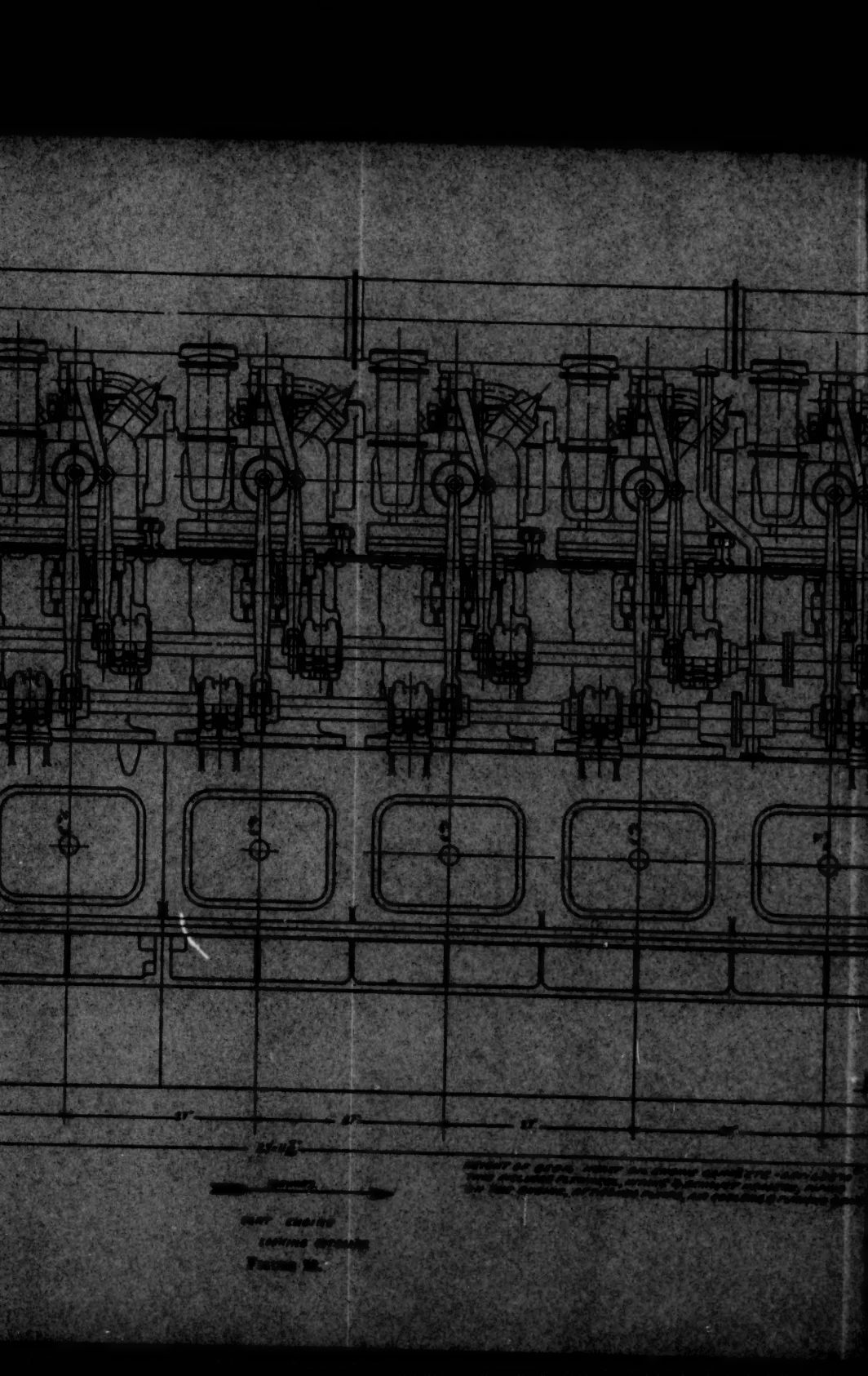
Both these sizes of engines passed acceptance shop tests and under the pressure of war large numbers of them were contracted for to engine the submarines then urgently needed.

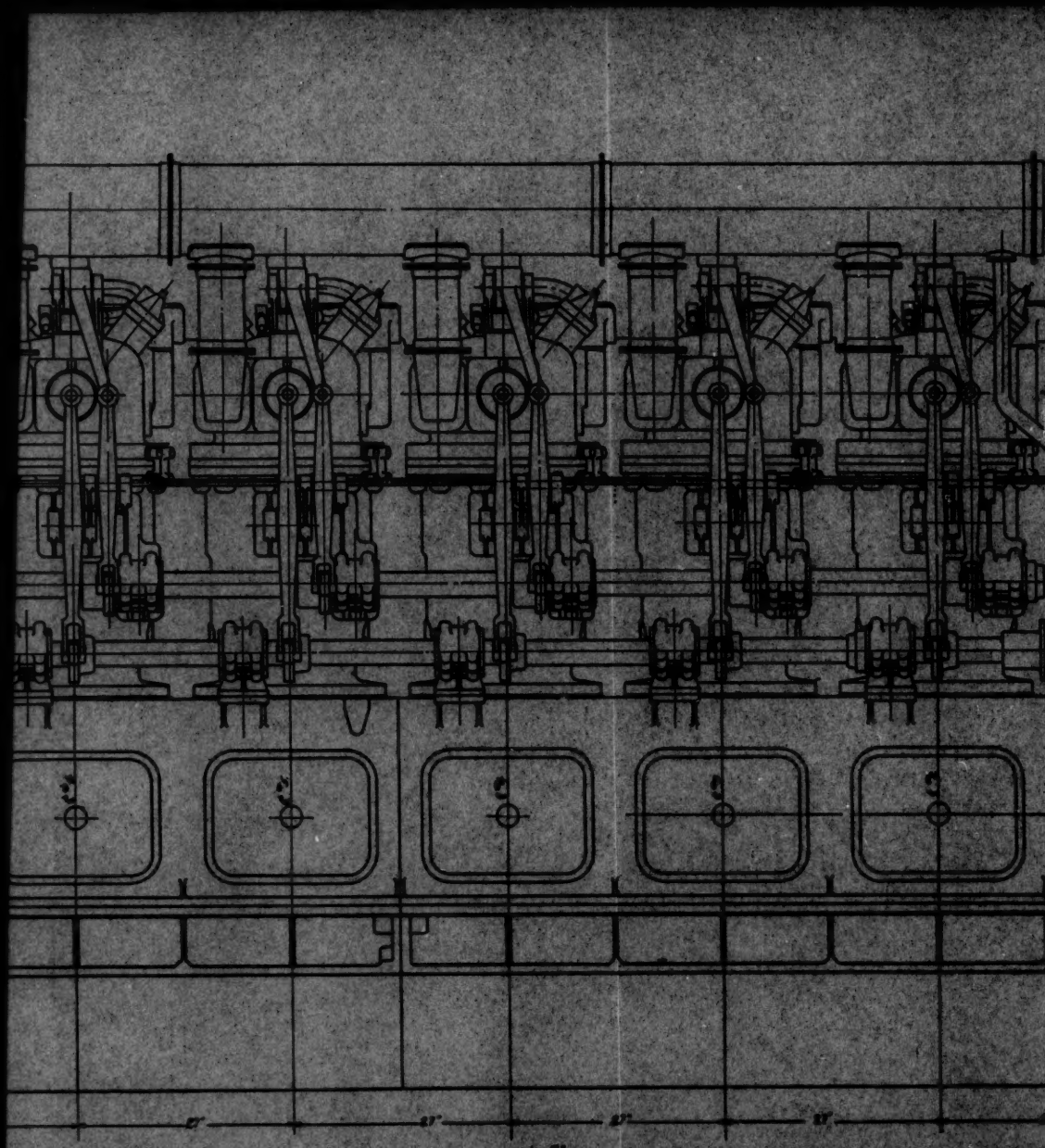
However, their duplicates reached the test blocks about the same time that the original engines were installed in the boats and put in operation, and then, both in the shop and in the boat, and in both sizes of engines, several crankshafts were broken due to synchronous torsional vibration. Space does not permit to go into the subject more deeply except probably to define the phenomena. In an oil engine torsional critical speed is the speed at which the number of explosion impulses in the cylinders per minute coincides with the natural period of vibration of the elastic shaft system, or even with any such part of it as $\frac{1}{1.5}$, $\frac{1}{2}$, $\frac{1}{2.5}$, $\frac{1}{3}$, the violence of them subsiding with the increase in the denominator of the fraction. The result of such a coincidence is an enormous increase in the angular deflection of the shaft and eventually a twisting break at the weakest section, an example of which is shown in Figure 24. The story of efforts to correct this weakness might as well be the subject of another paper—suffice it to say at this time that finally two different remedies were applied, the one by Nelseco was to increase the size of the shaft and thus displace the critical speed from the full power speed where it originally was, and the other used by the New York Navy Yard, sought by using a friction damper to decrease the danger, by preventing the vibration from developing.* Increased shaft diameter gave satisfactory results and a large number of Holland "S" boats are now in successful operation. There were three different dampers tried, which would give excellent results on shop tests and for a short time thereafter, but they would not stand up in service, and finally the speed of the engine had to be reduced.

As mentioned before, the bronze crankcase in the six cylinder engine gave trouble, therefore the final 8-cylinder, 4-cycle engine installed in the Holland "S" boats, received cast iron crankcase sufficiently stiff to take explosion stresses and also

* See paper by J. F. Fox in S. A. E. Journal, Nov. 1920, article by F. M. Lewis, JOURNAL AM. SOC. NAV. ENG., 1919, p. 857.


RELIEF VALVE
AIR SHUTTING VALVE
33'
27'
27'
27'





FRONT

FRONT ENGINES
LOOKING OUTBOARD

FIGURE 32.

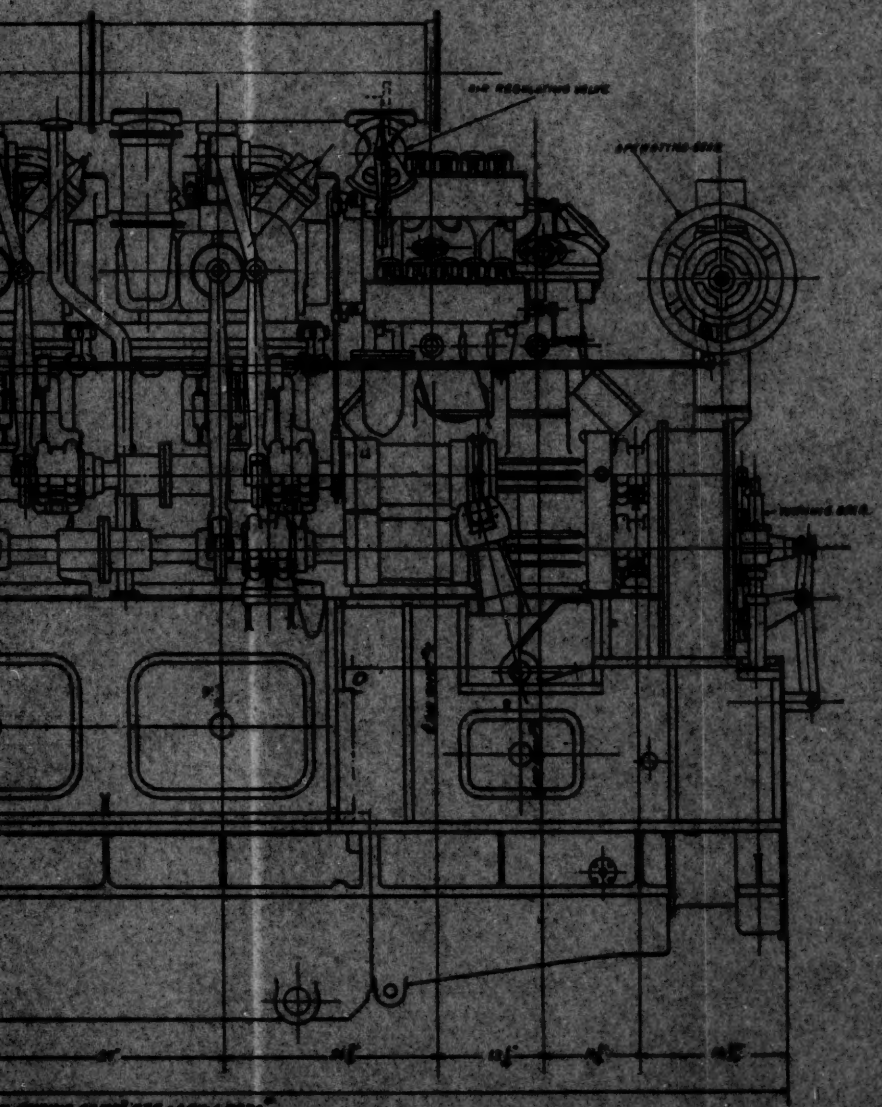


FIGURE 24.

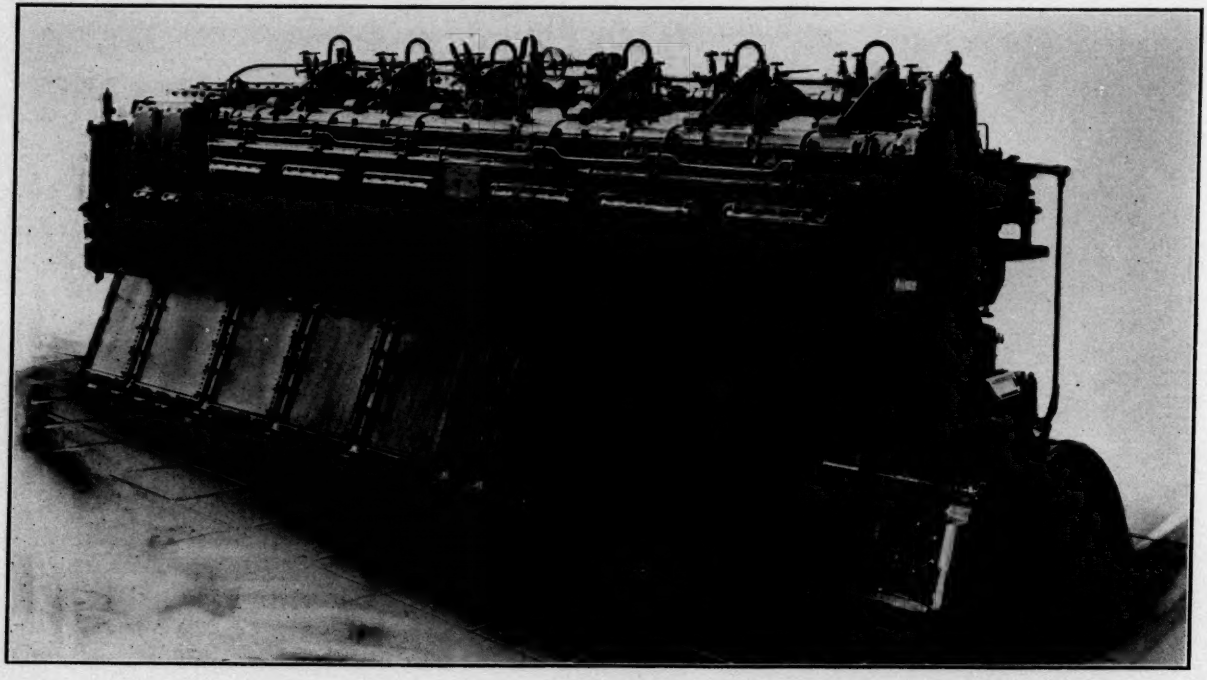


FIGURE 25.

cast iron bedplate. A three stage air compressor with two stepped pistons working in one cylinder block was also finally incorporated.

T–BOAT ENGINES.

As early as June 1914, Congress authorized, and soon thereafter the contract was signed for the building of a submarine capable of following the fleet at a rather high speed. This meant a powerful propelling plant for which as yet there was no precedent or example. In order to overcome this limitation the Electric Boat Company resorted to what proved to be a very dangerous arrangement, two six cylinder engines working in tandem on one shaft, keeping the twin screw type of drive. The war interfered and this development was set aside, and it was not until 1920 that the first boat of this type was commissioned.

The engines were substantially of the same type as the Nelseco 4-cycle, already described. They were designed to develop 1000 B.H.P. at 375 R.P.M., with working cylinders of 18-inches bore and 19-inches stroke, with oil cooled pistons and also equipped for both reversing and air starting.

The hulls of these boats proved very successful and despite their length they were easily handled while submerging, but the propelling plant had so many weaknesses that the boats could not be operated successfully for any length of time. The design of cylinder, cylinder head and piston did not come up to the requirements of high piston speed and heat developed in the cylinder and casualties were numerous. The two stage air compressor gave trouble and such compressors are not now being built. The engines vibrated badly causing numerous casualties, piston cooling mechanism suffering excessively. It is evident that two engines coupled together without any regard to the firing order of the cylinders would develop excessive vibration. The unevenness of wear in the two engines on one shaft caused clutch trouble. These experiences contributed

FIGURE 25.

to the development of electric drive in modern large submarines, where the forward engines drive generators and the main shaft through the main motors.

BUSCH-SULZER TWO-CYCLE ENGINES FOR THE "S" CLASS.

The Lake type *S-2* received a Busch–Sulzer two-cycle engine, the design of which is a development of the earlier engines for the "L" Class. The essential differences are in the working cylinder receiving a separate cylinder head; the rotary scavenging valve, instead of double seat poppet valve, controlling the upper scavenging slots; new type of so-called automatic shutter valves on the scavenging pump; flat disc valves on the air compressor. The engines are non-reversible, air starting, with both circulating water and lubricating oil pumps motor driven.

Figure 25 is an outside view of this engine.

These engines on shop tests gave rise to most optimistic predictions and installed in the boat made it one of the fastest submarines in the United States Navy. Soon after its commissioning, it developed clutch trouble which later calculations showed to be the result of torsional vibration. Kind fate interfered, a new and much heavier clutch was installed and the critical automatically passed out of the running range. At the time that she was scheduled to accompany several other "S" boats to the Philippines she had but little operation on the Atlantic but made the trip without much mishap and has done well at the beginning of her operation there. Being the only one of the type and a very fast boat, she has suffered more than other boats from the difficulty of obtaining competent personnel. These engines developed piston troubles.

Duplicate engines of this type were installed in four more submarines of the "S" type and operated for some time successfully until they demonstrated the ill effects of torsional vibration by breaking three shafts in close succession Calculations proved the critical speed to be at about 310 R.P.M.,

which was a favorite charging speed of the boats. Meanwhile we have learned not only to calculate in advance the critical speed but also obtain records of such vibration. Again two different ways were pursued—both based, however, on displacing the critical out of the operating range—one by adding elasticity in the shape of an elastic coupling, the other by stiffening the system with a larger crankshaft. Both proved satisfactory as subsequent torsiograms showed.

Piston trouble made itself felt here also and efforts to correct it by fitting forged steel piston heads are now in progress. Overloading as the result of carrying the auxiliary load of the ship, which in this case, due to detached pumps. is very large, probably contributed largely to the trouble and will be dealt with further in the concluding part of this paper.

POLAR AND CRAIG ENGINES.

During the war when the "E" and "F" Classes of submarines had to be re-engined, two more different types of Diesel engines were installed. *E-2* received an 8-cylinder, 4-cycle, non-air-starting and non-reversible engine of the Swedish *Polar* type and manufacture, furnished by McIntosh and Seymour, of Auburn, N. Y. The engine was designed to develop 250 B.H.P. at 375 R.P.M., with cylinder of 9-inches bore and 13-inches stroke. These engines were of the conventional valve in the head type and made a good submarine out of the *E-2* but at the end of the war the hull was scrapped and the engines installed at the Experiment Station at Annapolis.

F-3 was equipped with a 6-cylinder, 4-cycle, air-starting and reversible engine built by James Craig, E. & M. Works, of Jersey City, N. J. The feature of this engine was the auxiliary exhaust controlled by a valve at the end of the power stroke—a feature that proved quite successful on gasoline engines of the same manufacture. Apparently the only troublesome feature about this type of engine was its gravity feed to the main bearings.

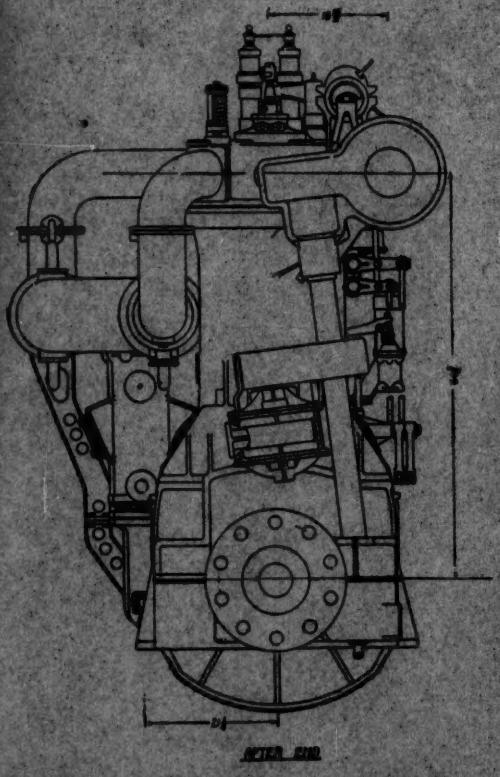
However, neither of these engines were ever duplicated or exercised any influence on the development of the Diesel engine for submarine service.

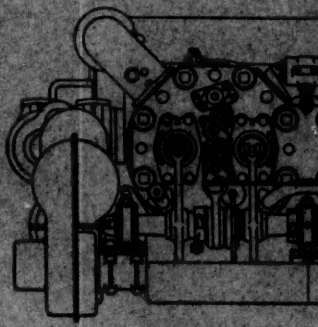
BUREAU TYPE 4-CYCLE ENGINES.

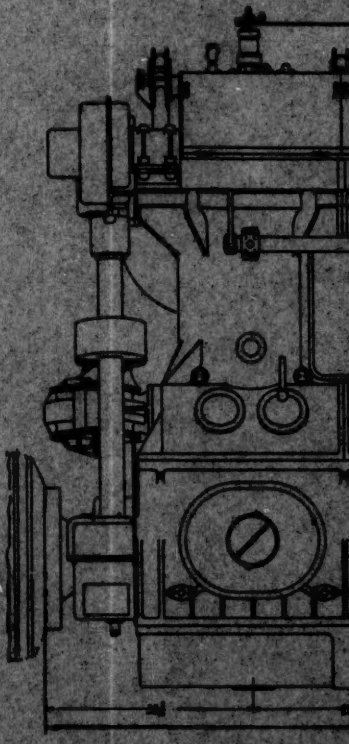
As the result of winning the war from Germany, all her submarines and engines built and in course of construction fell into the hands of the allies. United States crews brought across half a dozen German U-boats and operated them for a short time to study the state of the art. Besides the United States government purchased several large engines from the British. The German engines were the result of the combined efforts of the foremost Diesel engine builders of the country and the wide operating experience with the German submarines, and did not fail to impress very favorably our crews that brought them over by their reliability—a sure proof of perfection in design and workmanship. Accordingly the engines were taken out, subjected to various tests in the Navy Yards and were made the foundation of the Bureau type engines.

Figure 26 shows the general arrangement of this engine.

Several submarines of the "S" Class were equipped with a 6-cylinder engine developing 1000 B.H.P. at 425 R.P.M., with cylinders of 17¾-inches bore and 16-9/16 inches stroke. A four-stage air compressor driven by two cranks at the forward end and consisting of two differential pistons each having three stages is capable of furnishing ships air at 2500 pounds per square inch when required. Circulating water pumps of the single acting plunger type are driven from a cross shaft with four cranks, two of which serve to drive circulating water pumps and the other two drive the fuel measuring pumps. The lubricating oil pump of the gear type is driven either from the vertical shaft or direct connected to the foremost end of the main crankshaft. Their compactness, light weight, and general harmony of extremely complicated mass of mechanical devices depends entirely on the excellence







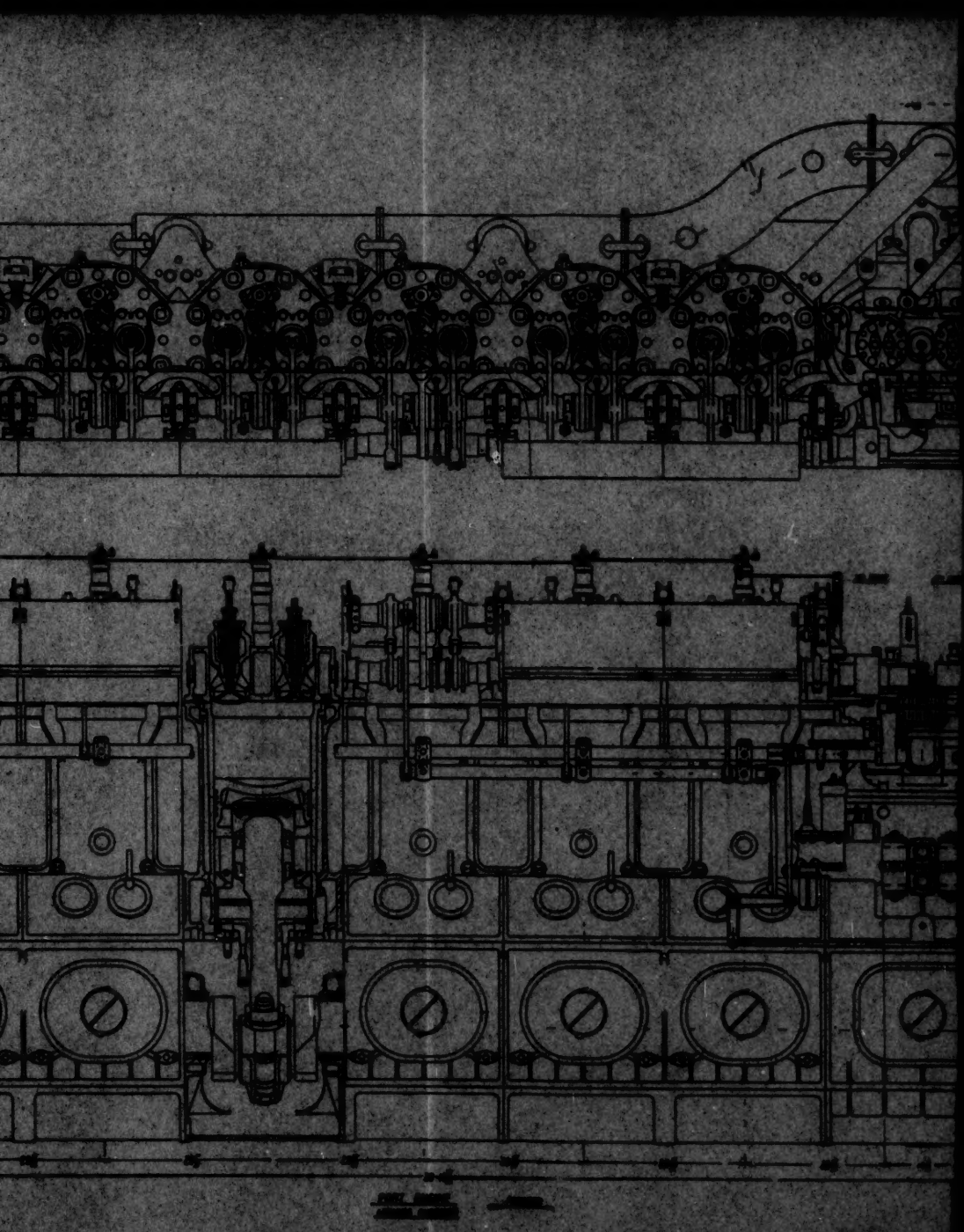


FIGURE 76.

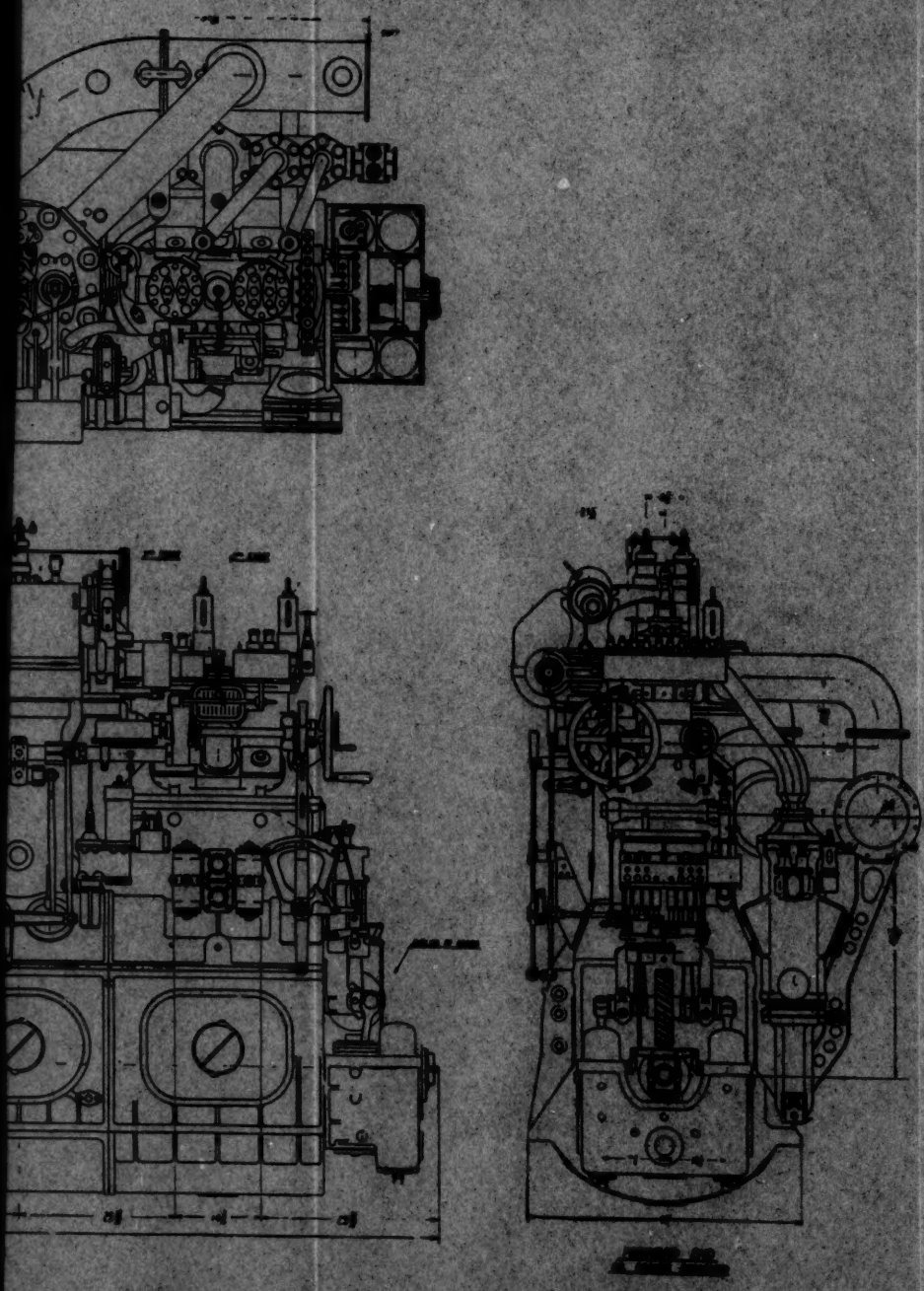
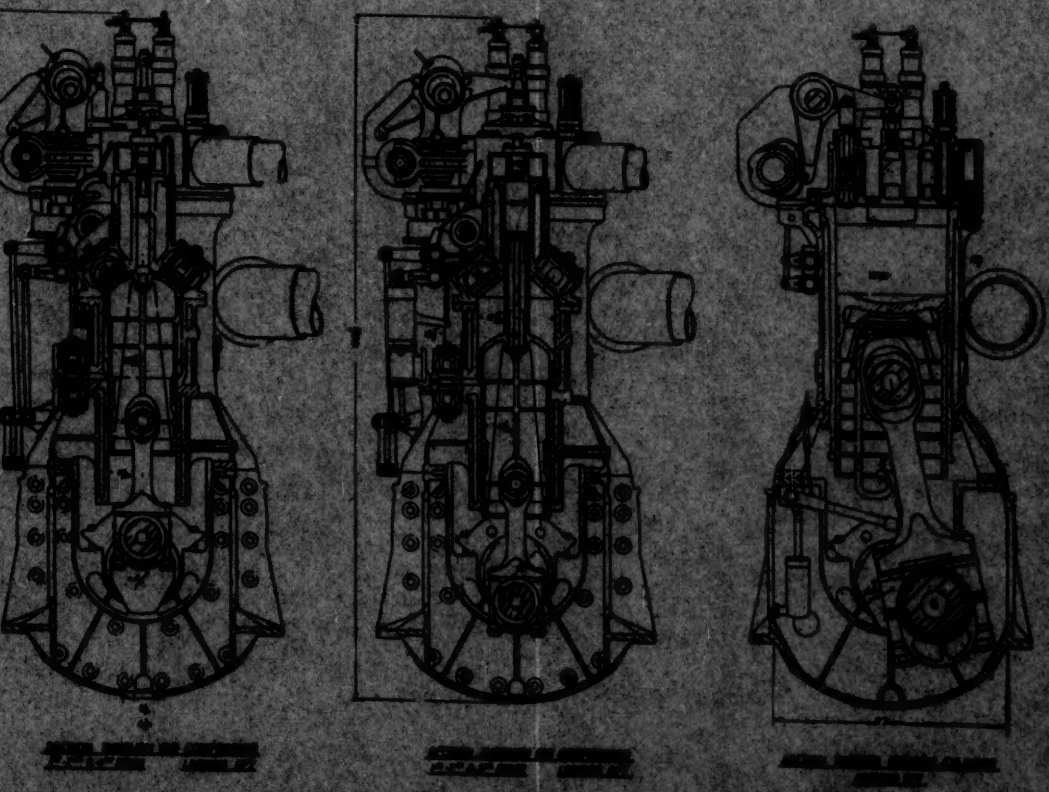


FIGURE 26.



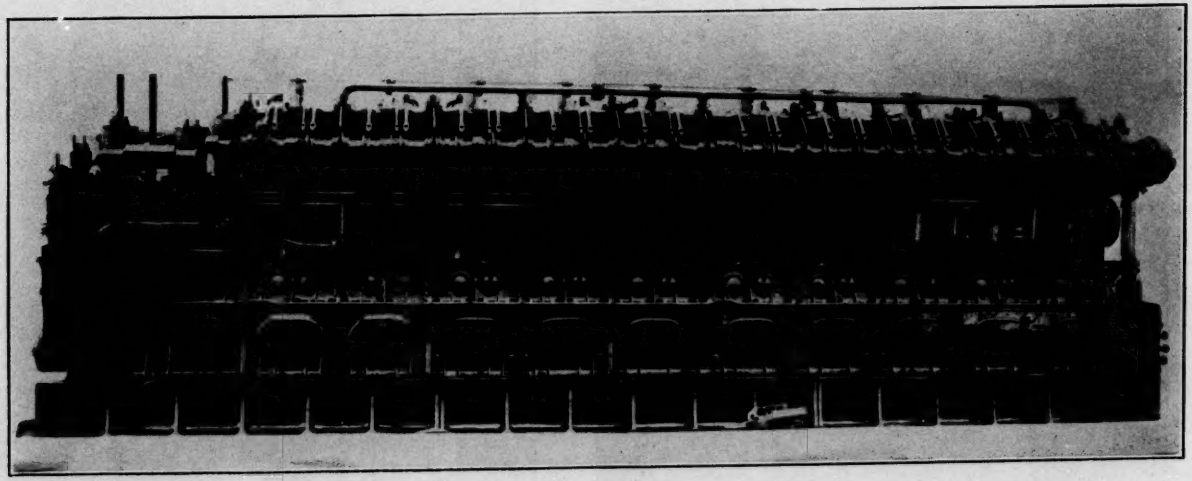


FIGURE 27.

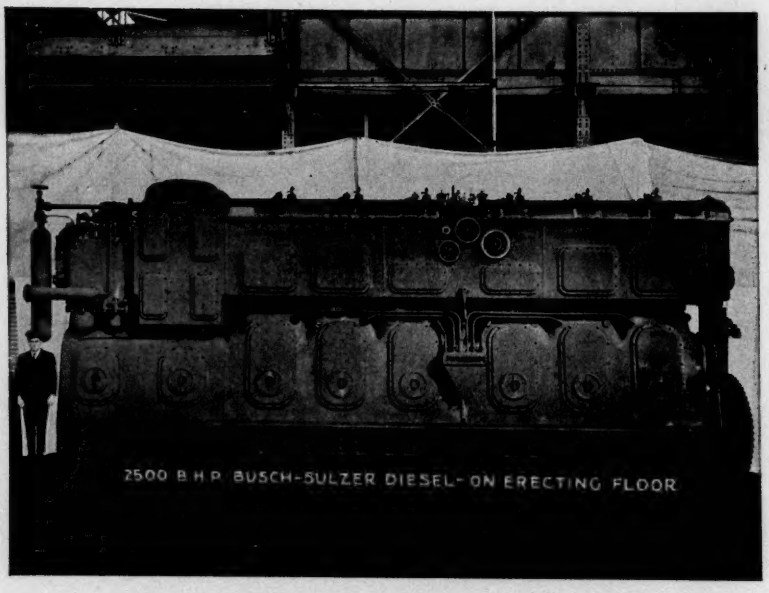


FIGURE 28.

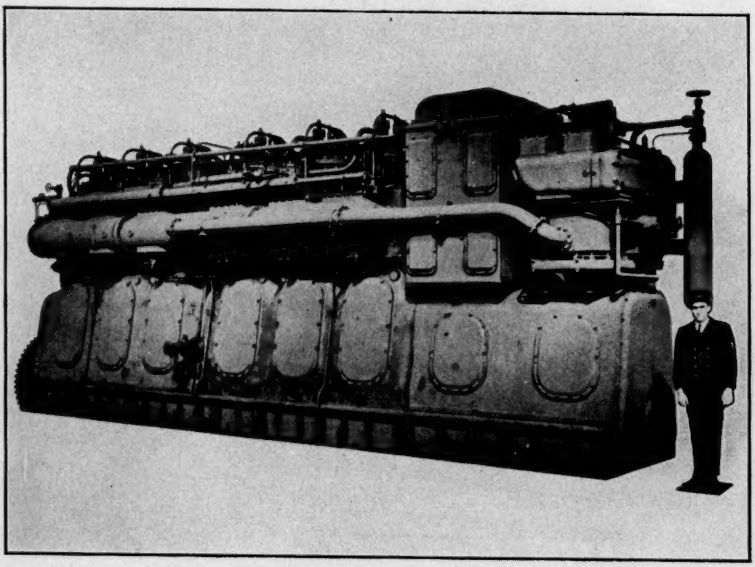


FIGURE 29.

of workmanship. Their design depends on the ability of a steel foundry to produce very difficult castings involving vast amount of experience or corresponding amount of scrap. No commercial foundry could be found to undertake the job and the government had to depend on its Navy Yards to do it. There are many interesting details about this engine which have been already widely published, the Navy Department even going so far as to distribute among reputable engine builders in this country complete sets of drawings of this engine in order to stimulate interest in Diesel engine building and give them the benefit of German experience bought by the lives of Americans lost in the World War.

Figure 27 shows a ten cylinder engine of this type, designed to develop 2350 B.H.P. at 345 R.P.M. Two German built engines of this type, purchased from the British are being installed in one of the "T" Class submarines and upon trials will become available as a power plant for future large submarines.

V-BOAT ENGINES.

This review would not be complete without the picture of an American designed and built engine that constitutes the main propelling plant of the "V" Class of submarines—A Busch–Sulzer two-cycle engine rated at 2250 B.H.P. at 310 R.P.M., with a maximum of 2500 B.H.P. at 325 R.P.M. Figures 28 and 29 show the inboard and outboard side of these engines. This engine is of the crosshead type and is direct reversible and equipped for air starting. These engines passed successful shop tests, and the results of their operation in service so far are most promising.

CONCLUSION.

The purpose of this story of successes and failures of Diesel engines in submarines is to point out the moral of the cooperation necessary between the three principals concerned in mak-

ing the submarine Diesel propelling plant a success—the engine builders, the boat builders, and the naval operating personnel, with the Bureau of Engineering as a liaison.

In the past the engine builder would design and build an engine to develop a certain horsepower at a certain rotative speed. The boat builder on the other hand contracted for a boat to have a certain speed on trials when right out of dry-dock with clean bottom and only in light trim while the commanding officer was expected to operate a military vessel fully equipped, and at all conditions of bottom, attempting to make contract speed without any regard to the limitations of the engine. There can be no two opinions on the weight of the engine installed—the scales determine that, its bore and stroke are also beyond dispute, but its horsepower is an arbitrary figure and probably no two men having anything to do with the engine can agree on that. With different types of engines the relation between the horsepower of the engine and the dimensions of its bore and stroke may be quite different and in turn the rated horsepower of the engine bears no fixed ratio to what an engine can pull for a short period of time or in an emergency. An engine rating, however, is governed by the amount of fuel that can consistently be burned in its working cylinders without injury to any of its parts exposed to the heat produced thereby. Obviously different designs of such parts exposed to heat can withstand different degrees of heat and it is up to the builder of engines to know the limitations of his design and up to the boat builder to provide means so that the operating personnel can comply with such requirement. The realization of the above cost all concerned much time, money, and countless disappointments.

Originally the submarines were operated on R.P.M., *i.e.*, by the speed of the propeller necessary for any given speed of the vessel, then again the engines were not to exceed their horsepower regardless of the speed of the shaft—in a submarine with a motor that is not less than the horsepower of the engine, it is very easily done. But now the engines are

protected by limiting the M.E.P.—mean effective pressure—which is a measure of the oil burned in the cylinder and the only index of reliability of the engine. The real reason for many a troublesome engine installation in the submarines of the past lies in this disregard for the necessity of limiting the M.E.P. developed.

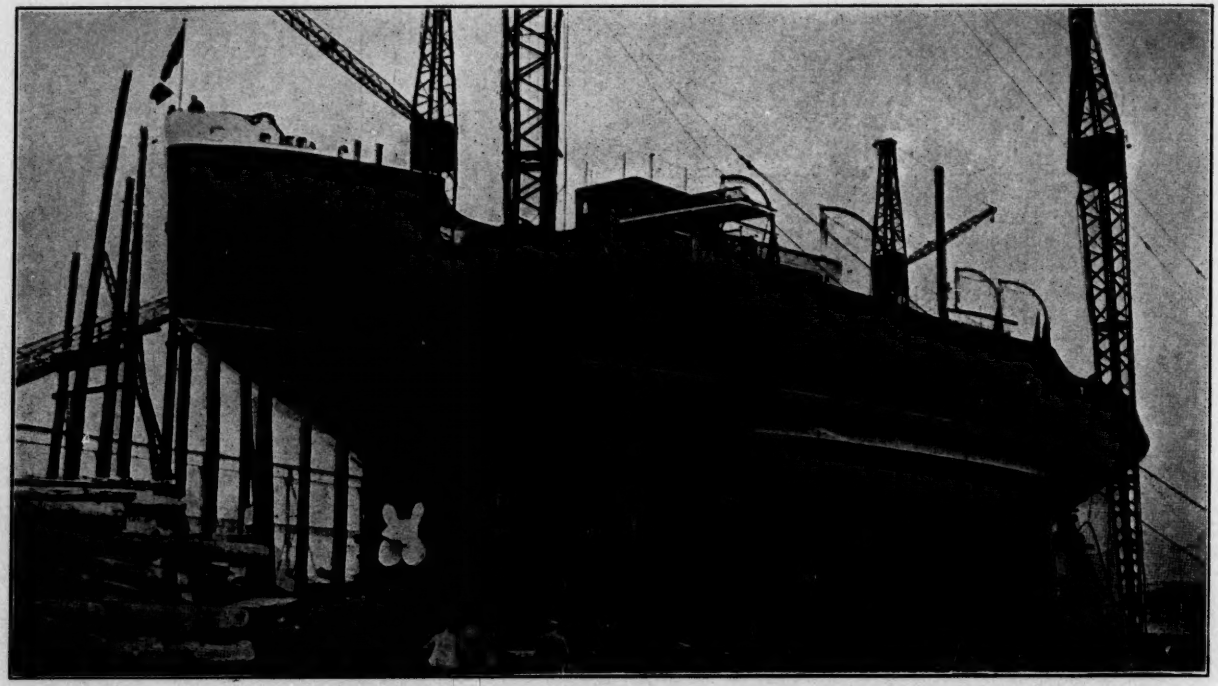
No less a glaring lack of teamwork between all parties concerned is the question of accessibility. The engine is built and tested with all of the room of a modern erecting shop around it and with an electric crane overhead, the boat runs trials with a brand new engine just tuned up and with a picked band of experts for a crew, but the commanding officer has to operate his boat in all kinds of weather with a continuously changing crew subject to orders from his superiors. It matters much to him whether the engine can be easily kept in good order or whether three days are necessary to examine air compressor piston rings with the result that such examinations will be put off until there is no more air coming from the compressor. Even the best of submarine propelling plants require much work and many sleepless nights from their operators and while the boat may be at the dock after a day's work somebody has to charge the batteries far into the night for the next day's operation and repairs must be made also. This often has to be done on engines installed with a shoehorn so that only the naked and perspiring body of a bluejacket can crawl through the recesses left between the engine and the skin of the ship.

The losses from the much advertised critical speeds of torsional vibration would have been very much less if the question of individual responsibility of three parties had not been present. It is true that lack of knowledge on this question should not be overlooked and the responsibility for the operation in critical speeds which were unknown to the operating personnel, and were left undetermined by either the boat or the engine builder, is difficult to place, and yet the blame reaches all three. In modern installation critical speeds are figured

out in advance and by suitable changes in various elements entering into the calculation, the more dangerous criticals placed out of the operating range. After the installation is completed, torsiograms are taken and the calculations are thus checked, and finally the operating personnel afloat is warned against running at undesirable speeds.

All these experiences were not in vain and the Navy realizes now that if success is to be achieved in the operation of Diesel engines—much study by all hands is necessary, and various means have been adopted to convey to those concerned the essentials about the engine itself and the installation generally. Such feats as a trip of several submarines to Alaska, accompanying the late President Harding, and lately a trip of a squadron of them to the Philippines with hardly a mishap, show the gratifying results of such a campaign of education.

In the last few years the state of the art in general has advanced to such an extent, and the operation of the Diesel engines in submarines are so satisfactory, that to predict an oil driven battleship instead of the oil fired one of today does not require extraordinary vision.



THE ICE-BREAKER "KRISJANIS VALDEMARS": BOW VIEW.

NOTES.

THE ICE-BREAKER "KRISJANIS VALDEMARS."

The ice-breaker *Krisjanis Valdemars*, which was launched on the 9th June by Messrs. William Beardmore & Co., Limited, at their Naval Construction Works, Dalmuir, to the order of the Marine Department of the Government of Latvia, will constitute a very important addition to the facilities of the port of Riga. The vessel is intended to operate in the Gulf of Riga, so as to ensure that the port will be accessible to shipping at all seasons of the year, and she has also been equipped for salvage purposes.

The following are the leading particulars of the ship:—

Length on waterline, feet and inches	185–0
Breadth on waterline, feet and inches	54–0
Depth moulded to upper deck, feet and inches	28–6
Normal draught, feet and inches	22–0
Normal displacement, tons (about)	2,800

GENERAL ARRANGEMENT.

As will be seen from the photograph reproduced, the ice-breaker has a specially designed stem with considerable rake under the waterline and a stern of cruiser type, both stem and stern-post being of cast steel, reinforced as necessary for ice-breaking purposes. The framing, which is on the deep-frame principle, is exceptionally strong, the spacing of the frames being 18 inches amidships, reduced to 12 inches at the ends of the hull. The shell plating is also much in excess of the usual requirements of the Bureau Veritas, with whom the vessel is to be specially classed. The ice belt from about 6 feet above the waterline to about 9 feet below is 1 inch in thickness fore and aft, and the end plating down to the keel line is practically of the same thickness.

In addition to the heavy, closely-spaced transverse watertight bulkheads, there are on each side of the vessel, for about three-fourths of the length amidships, longitudinal watertight bulkheads forming coal bunkers and water-ballast tanks. The whole system of subdivision is so arranged as to enable the ship to obtain the special mark assigned by the Bureau Veritas to vessels which are capable of remaining afloat with any two compartments open to the sea. By rapidly transferring water from the end tanks forward to the end tanks aft, or *vice versa*, the ship may be quickly trimmed either way. Similarly, by transferring water from tanks on one side of the vessel to those on the other, she may be rapidly heeled, and by these means she will be able to free herself when locked in ice.

Special arrangements are made for taking another ship in tow, a recess being provided in the stern of the ice-breaker so that the stem of the following ship may be drawn into it and held in that position by means of a large towing winch, of exceptionally powerful type, fitted on top of the after deckhouse.

Electric light is installed throughout, the generating plant consisting of two engines and dynamos each of 20 kw. and 120 volts. Wireless telegraphy, two flood-light projectors and a powerful searchlight projector are also fitted.

An electrically driven submersible salvage pump, with a capacity of about 200 tons per hour, is provided, with arrangements for transferring it to a vessel in tow or alongside so as to render assistance in case of damage.

The living quarters of the vessel include two large staterooms and a comfortable saloon for directors. The remaining accommodation throughout is ample and roomy; and the arrangements for heating and ventilating, and the insulation of the sides of the ship, bulkheads and decks in way of all living quarters, are such as will ensure the maximum comfort for the officers and crew compatible with the most trying climatic conditions which will be encountered on the service for which the vessel is intended.

PROPELLING MACHINERY.

The ship is provided with two sets of triple-expansion steam engines, which, together with the boilers, have been constructed by the shipbuilders. One set, for driving the after propeller, has cylinders 26¼, 42¼ and 69 inches diameter, by 43½ inches stroke; while the other set, working the propeller at the fore end of the vessel, has cylinders 16½, 26½ and 43½ inches diameter, by 20 inches stroke.

Steam is supplied to both sets of engines by four single-ended coal-burning boilers, each 15 feet 9 inches diameter by 11 feet 10 inches long, working under Howden's system of forced draught. Smokebox superheaters of the Sugden type are also fitted to provide a moderate amount of superheat.

The construction of the machinery throughout is very massive in character, in view of the extremely heavy stresses which will have to be borne on service. The shafting and propellers are of special nickel steel, which, together with the whole of the other steel materials—plates, bars, forgings and castings—for both the hull and machinery, have been supplied from the various departments of the shipbuilding company.

The principal functions of the after engine are to propel and maneuver the vessel, and it is capable of developing about 4000 I.H.P. for these purposes. The forward engine, driving the propeller at the fore end of the vessel, is used to assist in propulsion, and this engine is capable of developing about 1500 I.H.P. When working together, it is anticipated that the two sets of engines will give a speed of about 14½ knots.

The forward propeller, however, is primarily fitted to assist in ice-breaking, so that when the stem of the vessel is on the ice the propeller acts as a suction pump, sucking the water from under the ice and thus withdrawing its support. The weight of the ship, assisted by any water ballast transferred from the after to the forward end, will then break down the ice, enabling the vessel to forge ahead and clear a passage.

The very complete equipment of auxiliary machinery provided includes independent air pumps by Messrs. G. & J. Weir, Ltd., Glasgow; circulating pumps and forced-draught fans by Messrs. Matthew Paul & Co., Ltd., Dumbarton; feed pumps by Messrs. A. G. Mumford, Ltd., Colchester; and two large centrifugal pumps, each capable of discharging about 900 tons per hour against a head of about 50 feet, by Messrs. Henry Watson & Sons, Ltd., of Walker Gate, Newcastle-on-Tyne. The last-named pumps are used for transferring a large quantity of water ballast from the forward to the after tanks, or *vice versa*, and from tanks on one side of the ship to tanks on the other, in about 10 minutes.—"The Shipbuilder," July, 1925.

A SUGGESTED METHOD OF INCREASING THE EFFICIENCY OF THE SCREW PROPELLER.*

By Walter Pollock.

It is well known that the water flowing from a ship's propeller moves tangentially and axially as in the upper view in Figure 1. The object of the "Star" contra-propeller is to convert the tangential flow into an axial flow and so increase the propeller efficiency. This is effected by means of guide blades, which may be fixed either forward or aft of the propeller, or both. These blades are so formed as to divert the flow of water as it leaves the propeller blades in a direction more or less perpendicular to the face of the blade into a direction parallel to the line of the ship's course, as shown in the lower view of Figure 1. If, however, the contra-propeller is fitted forward of the main propeller, the water flowing through its blades has its direction changed before it reaches the main propeller, as shown in Figure 2.

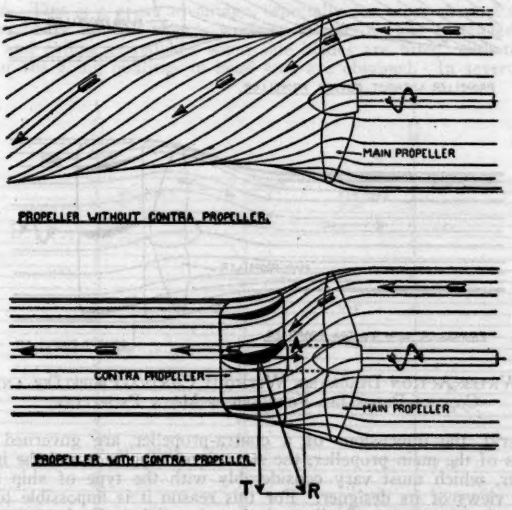


Fig. 1.—Comparative Water Action Diagrams Without and With Contra-Propeller Abaft the Main Propeller.

The blades are so curved that just sufficient change of direction is given to prepare for and to counteract the rotary motion ordinarily given to the water by the main propeller. Since the one change of direction is just neutralized by the other, the water is thrown off the main propeller blade in a direction parallel to the course of the ship. That the object aimed at has been achieved has now been proved, not only in experimental tank tests but in ships in actual service. The practical results obtained in actual service go to show that there is a very definite advantage to be gained. Hitherto, however, the variations among different types of ships in coefficient of dis-

* Abstract of paper read at the Spring Meetings of the Sixty-sixth Session of the Institution of Naval Architects, April 2.

placement, and in the formation of the stern frame and aperture for any given tonnage, have made it difficult to ensure a uniform gain in efficiency, but recent tank experiments have furnished information which will enable that difficulty to be overcome in the future. The "Star" contra-propeller can be built in a large number of forms, and a great deal of the success of an application will depend upon the correct selection of the type to be adopted, which must vary according to whether the vessel is an existing one or one to be built, whether she is to have twin screws, and so on.

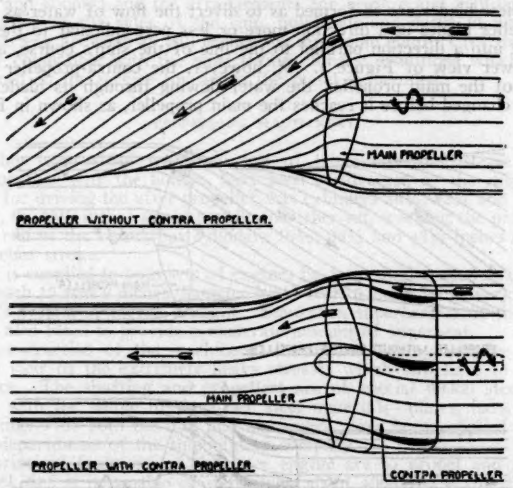


FIG. 2.—WATER ACTION DIAGRAMS WITHOUT CONTRA-PROPELLER AND WITH CONTRA-PROPELLER FORWARD OF MAIN PROPELLER.

In general, the dimensions of a contra-propeller are governed by the dimensions of the main propeller, the stern-frame aperture and the indicated horsepower, which must vary considerably with the type of ship and the particular views of its designer. For this reason it is impossible to standardize the device as regards dimensions and weight. Each case must be dealt with independently except where the vessel under consideration is a sister ship to one which is already fitted. In this case it might be possible to use the same drawings for the second vessel, making the necessary alterations in the process of fitting. The diameter of the contra-propeller may be taken to be approximately 55 per cent of the diameter of the main propeller. One of the chief objections which may be raised to the contra-propeller is the weight, and it is quite true that no shipowner would fit a device weighing several tons, as it may do in a large vessel, unless he could see some substantial advantage. It is therefore desirable to reduce the weight as much as possible so that the very slight decrease of the deadweight cargo capacity may be reduced to a minimum, and also that the stresses on the stern frame and stern plating may not become excessive. The weight is governed by two factors—the stern-frame aperture and the indicated horsepower. It is also obvious that, besides the weight to be carried, the contra-

propeller will offer additional resistance though this is not so great as would appear at first sight, because in most types the leading edges are very fine and the trailing edges well rounded; whereas in an ordinary stern frame the forward surface of the rudder post is flat, and in a 10,000-ton ship is often 8 inches wide and 21 feet high, which is equivalent to a flat surface of 14 square feet, while the after-side of the post has a flat surface of 12 square feet.

MINOR ADVANTAGES.

There are a number of minor advantages to be obtained by fitting a "Star" contra-propeller—for instance, improved steering is obtained, as will be seen by Figure 3, which is almost self-explanatory. The trials from which this diagram was prepared extended over a period of half-an-hour in a smooth sea, and the displacement was approximately the same in both cases. As rudder "bias" is almost entirely dispensed with, any single-screw vessel fitted with a contra-propeller will steer as easily to port as to starboard and *vice versa*. This is a great advantage, especially for cross-channel steamers, tugs, ferry-boats, and all vessels working in narrow and restricted waters. Pitching and rolling is reduced when side blades are fitted, with the result that better sea-going qualities in head seas are obtained. In several of the

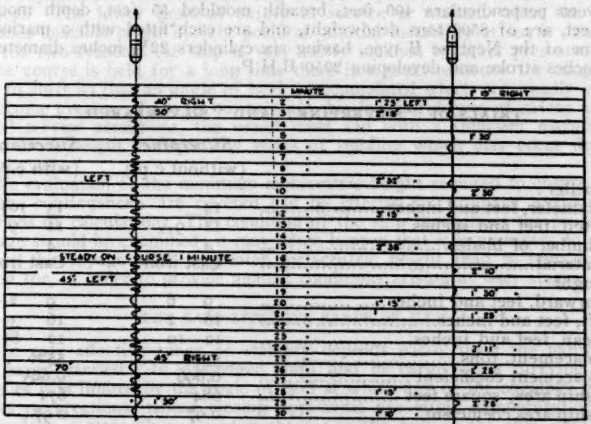


FIG. 3.—COMPARATIVE COURSE DIAGRAM.

vessels fitted with the device the advantage has been considerable, and in one case it has been stated that the increased speed in a heavy sea was one and a half knots. Vibration also is reduced as the water is less broken. It is unnecessary to emphasize the advantage of this in the reduction of wear and tear and the cost of upkeep and repairs to the main engine and shafting. Another point which is of very great importance in geared turbine vessels is that the torque variation in the shafting can be reduced by a contra-propeller. A propeller blade, in passing the rudder-post, is subject to a reactionary thrust which momentarily increases the torque in the propeller shaft. With the propeller revolving at a constant speed, there is, therefore,

a regular increase and decrease in thrust, and consequently in the torque to which the shaft is subject. Since the reduction gear has an even torque transmitted to it by a very finely balanced turbine rotor, and is working against a regularly varying load in the shafting, stresses are set up which result in wear of the gear and teeth. The greater the number of impulses per revolution the more even will be the torque on the shaft, since the increase of load due to one impulse is balanced by the decrease of load of the impulse immediately before it. Further, the blades of the contra-propeller are shaped in such a way as to receive the water without shock. This reduces the intensity of the impulses transmitted to the main propeller, so that the effect of fitting a contra-propeller is not only to reduce the variation in load but also to reduce the maxima of the impulses transmitted due to this effect. Lastly, several of the types with side blades are a distinct protection to the main propeller against damage by small craft or any floating or partially submerged objects as the obstruction would be met by the fixed contra-propeller instead of by the main propeller, and the ship could still continue her voyage in a satisfactory condition, whereas the consequences of damage to the main propeller are too well known to need elaboration.

The following results will give a good idea of the extent to which the efficiency of several vessels has been increased by fitting a contra-propeller. The *Silverpine* and *Silverlarch*, which were built in 1924, have a length between perpendiculars 400 feet, breadth moulded 55 feet, depth moulded 38 feet, are of 8500 tons deadweight, and are each fitted with a marine oil engine of the Neptune B type, having six cylinders 22½ inches diameter by 45 inches stroke and developing 2250 B.H.P.

TRIALS OF "SILVERPINE" AND "SILVERLARCH."

	Silverpine (without c.p.)	*Silverlarch* (with c.p.)
Propeller:		
Diameter, feet and inches	14	13 10½
Pitch, feet and inches	11 9$\frac{5}{16}$	11 9
Number of blades	4	4
Material	Cast iron	Cast iron
Draught:		
Forward, feet and inches	9 6	9 2½
Aft, feet and inches	16 2	16 3½
Mean, feet and inches	12 10	12 8½
Displacement, tons	5576	5526
Displacement coefficient	0.696	0.695
Midship area, square feet	684	671
Midship area coefficient	0.97	0.97
Propeller tips immersed, inches	13¾	15¾
Speed, knots	11.11	11.85
Slip, per cent	9.4	4.8
Revolutions per minute	105.4	107.25
Brake horsepower	2070	2085

The *Silverpine* ran her trials without the "Star," and the *Silverlarch* ran trials with the "Star," in order that the comparison might be made. In this case the trials were the usual light draught trials carried out by the builders on completion. The results of voyages on actual service are expected to show a still greater improvement. From the figures given it will be seen that an increase of 6.66 per cent in speed was obtained for practically the same brake horsepower. The decrease of 4.6 per cent in the slip is typical of the increased propulsive efficiency of a well-designed contra-propeller,

and as the trials were run over the same measured mile under similar weather conditions, they may be taken as giving a very fair comparison of results with and without a "Star."—"Shipbuilding and Shipping Record," April 16, 1925.

STABILITY AND SEAWORTHINESS.*

By C. Frodsham Holt, M. Eng.

When a ship proceeds to sea upon any given voyage it is very rarely that the metacentric height is known with any degree of accuracy. Even if the actual metacentric height is known it cannot be said from inspection whether the stability is and will remain satisfactory during the voyage. Before attempting to discuss how best to give information to the shipmaster regarding the safety of his ship it is necessary to decide what constitutes safe stability.

The first condition essential to seaworthiness is that the ship shall never be unstable when upright at the beginning or the end of any given voyage or while at sea. The second condition of seaworthiness is that the worst conjunction of wind and waves that can possibly be encountered during the given voyage shall not cause the ship to roll beyond the safe effective range of heel. The lower limit of safety in service is reached when the ship becomes unstable in the upright position. Initial instability at sea is dangerous for two reasons, the ship is very easily inclined by wind pressure, and if one course is held for a long time there is a great tendency for cargo or coal to shift so that an angle of heel is accumulated which continually tends to become greater. In order to maintain this limiting degree of safety under actual service conditions, the vessel must sail with a sufficient margin of metacentric height to cover any losses of stability which may occur during the voyage.

The evaluation of the minimum metacentric height required to ensure that all the contingencies are covered can be effected by ordinary well-known methods of calculation. The condition of the ship throughout the projected voyage should be examined; it may often happen that it is necessary to commence the voyage with a greater metacentric height than is immediately called for, in order to provide against conditions arising at later stages.

THE SAFE EFFECTIVE RANGE OF HEEL.

The safe effective range of stability depends partly upon the nature of the cargo carried. In passenger ships and in cargo ships carrying bale cargoes it is limited by the angle of heel at which any of the openings in the decks or erections, whose closing appliances are either temporary or not equivalent in strength and watertightness to the hull of the ship, enter the water and become liable to broaching under the direct blows of storm waves. The importance of the security of openings in erections has been recognized by the Load Line Report of 1916, in which the freeboard assigned is made to depend on the type of closing appliances fitted. In ships carrying cargoes in bulk, the safe range of heel is alternatively limited to the virtual angle of repose of the cargo; the statical angle of repose is altered by the forces arising from the rolling of the ship. There are methods of estimating the angle of heel rolled to by the ship; but there is no standard method of calculation such as is used in making statical stability calculations, nor is there any recognized basis for assessing the worst weather conditions encountered by the ship.

* Abstract of Paper read at the Spring Meetings of the Sixty-sixth Session of the Institution of Naval Architects, April 3.

In view of the investigations of the late Dr. R. E. Froude† and of the fact that all ships, except those of very long periods, are liable to find waves with which synchronism can take place, it may be taken as true that if the safe effective range of heel is not exceeded under conditions of synchronism with the waves, it will not be so under any other condition. Thus the standard condition for rolling among waves to be considered in discussing the seaworthiness of the ship can be taken as that arising from synchronism of the ship's period with that of waves having the maximum wave-slope. In order to determine the maximum angle of roll it is necessary to know the resistance to the rolling of the ship.

RESISTANCE TO ROLLING.

The present state of knowledge respecting the resistance experienced to the rolling of ships is far from satisfactory. Not only are very little data available, but those best qualified to speak on the subject throw doubt upon the sufficiency of the accepted method of presenting the results in the two-power decremental equation.‡ As some knowledge of the resistance to rolling is essential if a standard of safe rolling is sought, this presents the chief point in the argument which is open to criticism. If this method should ever be adopted, it would prove useful to carry out systematic model experiments on merchant-ship forms in order to determine coefficients of resistance; the value of data so obtained would far outweigh the expense entailed. Nevertheless, the method holds good, even though there are little data available.

From the results obtained at the inclining experiment the inertia of the ship in any desired condition can be built up and the rolling period calculated. It is found from a number of calculations that the inertia of a given ship at a given draught varies very slightly with the stowage of the cargo; the change in the period is accounted for almost entirely by the alteration in metacentric height. Thus, unless for very exceptional circumstances, draught and *either* metacentric height *or* period together afford a sufficient index of the dynamical characteristics of the ship.

WIND AND WAVES.

It becomes necessary to examine the prevalence of weather conditions dangerous to the ship, and the degree of severity of their incidence. We are chiefly interested in the maximum wind pressure that is experienced at sea; but a certain interest attaches to the prevalence of the worst conditions, and to the degree of regularity of the wind during storms. A sea-going ship must keep the sea in all weathers; she must, therefore, be able to withstand the maximum wind pressure even when rolling most severely. It is suggested that, on the basis of actual records, any sea-going ship should be proved capable of withstanding a standard wind pressure of 55 pounds per square foot. The influence of the land on the maximum pressure is very marked; it seems reasonable to allow that ships engaged in harbors and estuaries will never need to withstand a greater wind pressure than 30 pounds per square foot. Again, cross-channel steamers, and others making only short runs in the open sea, are not allowed to run during extreme weather conditions, these ships, therefore, need not be required to withstand such severe wind pressure as sea-going ships. Nevertheless, they may be caught by unexpected bad weather. It seems possible to allow that they will be safe if they can withstand a wind pressure of 40 pounds per square foot.

† "The Non-uniform Rolling of Ships," "Trans. I. N. A.," 1896.
‡ "Rolling of Lightships," Baker and Idle, "Trans. I. N. A.," 1912.

The important phenomena of wave-motion that require notice in dealing with the rolling of ships are the regularity in point of size and period of successive waves, the prevalence of waves of periods with which synchronism is possible, and the maximum wave-slopes associated with waves of any given period. Scientific observations of the maximum dimensions of storm waves at sea are not numerous, nor are they always to be taken at their face value, as a regularly organized survey has never been carried out all over the world; those available arise only from chance conjunction of opportunity and circumstances. Reliance is placed here upon the published work of Dr. Vaughan Cornish ("Waves of the Sea and Other Water Waves") and Mr. Idle ("The Engineer," April, 1911). From Dr. Cornish's and Mr. Idle's observations the author has constructed a series of diagrams, on a base of rolling period of ship in synchronism with the waves, giving the maximum effective wave-slope which a ship of given period can encounter in service under the worst conditions of weather. From these the maximum possible angle of forced rolling can be calculated directly. In these diagrams are also plotted curves showing the effect of course and speed of the ship in modifying the apparent wave-slope and apparent period of the waves; for if the period of the ship lies beyond the possible range of synchronism applicable to her service, synchronism may yet arise when steaming with a fair wind and athwart the seas. The method of calculating the angle of heel produced by a given intensity of wind pressure is well known.

SAFE ROLLING.

If the foregoing calculation is performed for a ship at a given displacement, under the standard conditions of wind and waves, over a range of metacentric height and period of rolling, a series of maximum angles of heel is obtained, which can be plotted on a base of metacentric height. The curve so obtained crosses the safe range of heel appropriate to the given draught at two points, in general. Where the curve lies below the safe range of heel, the ship will never reach the limit of safety under any conditions at sea. The points of intersection give values of maximum and minimum safe metacentric heights for the given draught.

Prof. Jenkins, in 1887, read a paper before this Institution on "Shifting Cargoes." In it he showed that at any point in a free surface of cargo above the center of gravity the angle of repose is virtually reduced by the acceleration due to rolling motion. If, therefore, a free surface forms through the settling of the cargo, shifting will certainly take place at much smaller angles of heel than are reached if the ship is subjected to synchronous rolling. In a ship whose period is 4 seconds, a free surface of bulk cargo is situated 10 feet above the center of gravity, the maximum angle of synchronous rolling is 42 degrees. If the cargo is of grain having a statical angle of repose of 21½ degrees, the cargo will begin to shift in the wings at a radial distance of 20 feet from the center of gravity if the ship rolls to 16½ degrees. If the period of the ship were 8 seconds, the cargo would not begin to shift until an angle of roll of 20 degrees was reached, while the maximum angle of synchronous rolling is reduced to 34 degrees. If instead of grain coal having an angle of repose of 35 degrees were carried, the cargo would begin to shift when the angle of rolling was 25½ degrees in the 4-second ship and 32½ degrees in the 8-second ship; the maximum angle of synchronous rolling is the same as before. In order to eliminate, as far as possible, any chance of the cargo shifting, it should be arranged that any ship carrying bulk cargo at a higher level than the center of gravity of the ship will not be at sea with a rolling period within the range of possible synchronism.

STANDARD OF SAFETY.

In this way a standard of safety applicable to any ship has been arrived at. If a series of curves are drawn up for any given ship it can be said that under no circumstances will the stability be unsafe from the beginning to the end of any voyage commenced with a metacentric height lying between the maximum and minimum values appropriate to the draught at the time of sailing. There are three points at which criticism may be directed. First, it may be said that no sailor would allow his ship to come into synchronism with the waves in such extreme weather conditions as are here dealt with. It is true that he would not, if he could help it; it can always be avoided by altering the course. Yet machinery, propellers, and rudders do break down in bad weather; sometimes the instant for "heaving to" is badly chosen; occasionally the ship refuses to come up into the wind and hangs for a long time in a dangerous position. Again, dangerous rolling develops with great rapidity under conditions of synchronism; it may not be distinguished from bad non-synchronous rolling until too late. Secondly, it may be said that our present knowledge of the resistance coefficients which must be used is not sufficient. This is so, but it can be remedied very simply in a short time by making experiments, and at no very great expense. Thirdly, the effect of free water or oil fuel on the maximum angle of rolling has been neglected. Although the action of free water would seem to render the ultimate safety of the ship more secure, yet the possibility of unforeseen combinations of the periods of the various tanks giving the ship sudden violent lurches when synchronism is not present, makes it inadvisable to increase the permissible range of metacentric height.

PRACTICAL APPLICATION OF STANDARD OF SAFETY.

Having defined a standard of safe stability applicable to any ship the next stage is to apply it in such a way that the master can assure himself that his ship, at the time of sailing, conforms with the standard and will remain so throughout the voyage. There are two methods of doing this—indirect and direct. The indirect method consists in making more or less approximate estimates of the height of the center of gravity or of the metacentric height. Such estimates can be made without calculations by mechanical means, or by the use of tables. These methods depend, however, for the accuracy of their findings on the accuracy with which their user has taken account of the condition of his ship, and are not conclusive evidence that the ship actually sails with the desired amount of stability. If some item is overlooked, the presence of free water is not realized, or the cargo is not actually stowed as has been allowed, the result is affected, and the safety of the ship is not assured.

The direct method consists in testing the stability of the ship. There are two ways in which this may be done: (1) By inclining the ship and determining the metacentric height; (2) by observing the rolling period. The former method is well known, but requires time which can ill be spared, as well as some technical experience. The latter method can be made use of by means of the stability meter, an instrument devised for this purpose, with which the author has recently been experimenting. This instrument makes a record of the small oscillations performed by a ship in the comparatively still waters of a dock or harbor. It is calibrated during an inclining experiment, and a stability scale is then prepared from which the frequency of rolling at any time can be converted by inspection into metacentric height. It has been found to give results very close to those of an inclining experiment. For practical use on board ship stability scales can be made which show by inspection, after the frequency of rolling has been counted from a stability meter record: (1) The metacentric height at the

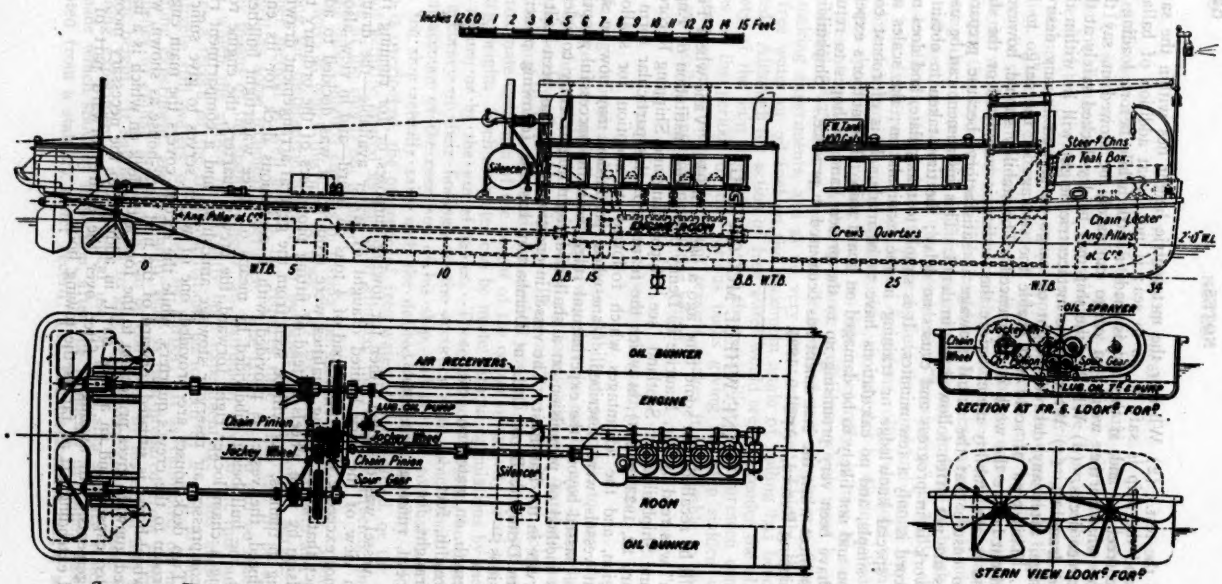
time of the test. (2) Whether the metacentric height lies within the safe range, and the ship is safe to go to sea; if not, what addition of ballast would correct it; and, if the test is made a little while before loading is complete, what further weight of cargo can safely be stowed in, say the upper 'tween decks. (3) Whether the ship may be dry-docked safely at the observed draughts. (4) Whether the metacentric height will lie within the safe range, after any desired weight of cargo is stowed in any desired manner. This type of scale will enable the stowage of the cargo to be schemed out before the work is commenced. The stability should, however, be tested by the stability meter before the loading is completed or the ship sails, to ensure that the actual stowage is satisfactory, because prepared schemes are not often followed out in detail. This instrument can be used while work is in progress and causes no delay; the time taken to obtain a new record is only a few minutes. It is simple to manipulate, and does not require special knowledge or training in the manipulation; the scales are equally simple, and no calculations have to be made. It is of robust construction and not likely to be damaged on board ship. The author's experiments have been very promising up to the present, and he hopes to return to the subject when the instrument has been used in practice.—"Shipbuilding and Shipping Record," April 9, 1925.

VANE-WHEEL MOTOR TUG.

It will be recalled that some time ago a paper entitled "Vane-wheel Propulsion" was read by Mr. Maurice E. Denny before the Institution of Engineers and Shipbuilders in Scotland (see "Shipbuilding and Shipping Record," February 22, 1923, p. 233), in which the features of this particular form of propulsion, and the advantages which followed its adoption for shallow-draught vessels, were described. Vane-wheel propulsion may now be said to have passed beyond the experimental stage, with the successful adoption of this method of propulsion on a shallow-draught tug recently completed for service in the Far East. The vessel in question, which has been built by William Denny & Bros., Ltd., at Dumbarton, has the following principal dimensions:—

Length, on waterline, feet	60
Breadth, feet	14
Draught, feet	2
Speed, running free, knots	9

The vessel was designed either for towing purposes or for running free, and in view of the very limited draught of water available—the draught must not exceed 2 feet with a load of 4 tons on board—and in view also of the fact that good towing qualities were required, it was decided to adopt propulsion by vane wheels instead of fitting propellers of the ordinary type running in tunnels. As will be seen from the general arrangement drawings reproduced, the vessel is provided with a continuous deck for its entire length, the hull being subdivided by means of three watertight bulkheads, yielding a chain locker right forward, the crew's quarters, the engine room with compressed air reservoir stowage amidships, and a compartment right aft. Two deck-houses are provided, one of which serves to give sufficient head room to the crew's quarters, while the second covers the main engine. The wheel-house is situated forward of the first deck-house as shown, while a raised gunwale gives protection to the forecastle deck on which is a hand-operated capstan and an anchor davit, in addition to the necessary mooring arrangements. An awning is fitted over the entire forward part of the vessel extending aft as far as the towing hook.



SHALLOW-DRAUGHT VANE-WHEEL PROPELLED MOTOR TUG. BUILT BY WILLIAM DENNY & BROS., DUMBARTON.

The most interesting feature of this little vessel lies, of course, in the method of propulsion. Two vane-wheels, each 6 feet in diameter, have been fitted, and these are driven through chain gearing from a single direct reversible heavy-oil engine. The engine, which was supplied by Norris, Henty & Gardners Limited, of Patricroft, near Manchester, has four cylinders and develops 96 B.H.P. at a speed of 370 R.P.M. when developing its full power. Owing to the size of the vane-wheels, the propeller shafts and the after bearings are well above the water line, and to connect these with the main shaft of the engine, chain gearing arranged as shown in the sectional drawing, also reproduced opposite, is employed. The chains, which are of Hans Renold's type, work in an oil-tight gear case, lubricating oil being sprayed on to the driving pinions under pressure by a pump driven from one of the chain wheel spindles.

The gearing gives a ratio of reduction of about 3.5 to 1, the vane-wheels having a speed at full load of about 105 R.P.M. In previous vessels fitted with vane-wheel propulsion each wheel had its own engine, but the highly satisfactory results obtained on trial have fully justified the arrangement adopted on this vessel, the gears being particularly quiet when running.

TRIAL RESULTS.

The vessel recently underwent an exhaustive series of trials on the River Clyde, both when running free and when towing, and during these trials the power developed, and the fuel consumption and the speed, were carefully measured. When running free a mean speed of about 9 knots was obtained, and when towing another vessel over 80 feet long the mean speed was about 6½ knots. The engine was run on crude oil, and the consumption during the trials was about 6 gallons per hour. It will be realized, therefore, that the cost of fuel for running this vessel is remarkably small.

The vessel has now been dismantled and shipped to her destination in the Far East, it having been found necessary to cut the hull in two in order to facilitate shipment.—"Shipbuilding and Shipping Record," June 4, 1925.

RETURNING EXHAUST STEAM TO THE BOILER.

In the "Engineer," May 29, 1925, C. O. Towler advances a new method of returning the exhaust fluid to the boiler without resorting to a condenser. He states that several attempts have been made, mainly in the remote past, to effect a saving in fuel in steam-power plants by some such process. It would be superfluous to discuss the reason for their failures, as in the majority of cases the exhaust steam was intended to be returned by means of either a pump or an injector, the injector being supplied by live steam from the boiler itself or from a supplementary boiler.

If a low-pressure fluid is to be admitted freely into a high-pressure system, there must exist in that high-pressure system a portion constantly maintained at a pressure below that of the low-pressure fluid. Such conditions are attained in practice by means of the Venturi tube in which the pressure energy of a fluid is partly converted into kinetic energy with a consequent drop in pressure.

It was stated that this system was successfully applied to an experimental steam boiler, and a portion of the boiler was maintained at a pressure lower than that at which steam was generated. These results were obtained by using short tubes closed at one end in place of the usual boiler tubes open at each end.

Using a short glass closed-ended tube, it has been observed that until the boiling point is reached, a circulation is maintained, water flowing in at the

bottom and out along the top of the tube, as indicated by the arrows in Figure 1. As soon as the boiling point is reached, however, bubbles of steam begin to form along the bottom of the tube. These bubbles steadily grow until practically the whole of the tube is filled with steam; then the water in the container forces out the steam, the tube becoming again filled with water, and the operation is repeated. Under more intense heat the rapidity of the operation increases, and eventually takes the form of vigorous pulsations, the water flowing in and the steam flowing out.

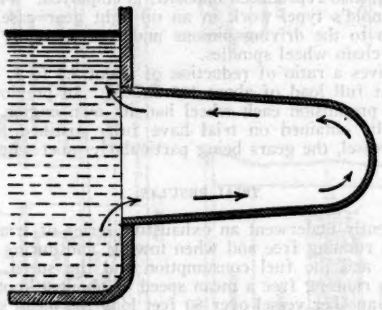


FIG. 1.—ACTION IN A CLOSED TUBE.

By restricting the orifice of the tube and thus giving the water and steam a greater velocity at this part, a section of the tube is maintained at a pressure lower than that at which steam is generated. The apparatus used to verify the foregoing supposition is illustrated in Figure 2. While the pressure in the boiler was 40 pounds per square inch, that at the restricted portion of the tube was 25 pounds per square inch, giving a pressure difference of 15 pounds. As the flow in the tube reverses twice per pulsation, the velocity of water and steam is not constant, and the pressure difference must be a mean value, although no fluctuation on the gage was discernible.

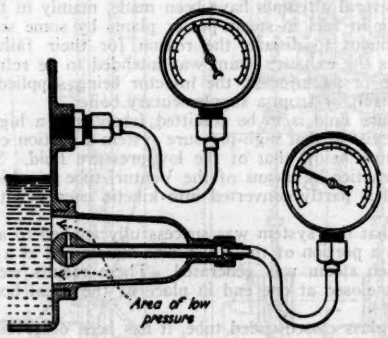


FIG. 2.—VENTURI EFFECT OF TUBE THROAT.

In applying the principle to return exhaust steam to the boiler, if all the exhaust steam is to be returned, in order to maintain the pressure in the boiler constant, a tube must be designed so that the amount of heat taken in per unit time by the tube is equal to the difference between the total heat of the quantity of steam taken in during the same period at the pressure of the boiler and the total heat of the same steam before entering. This neglects radiation losses and assumes that all the heat taken in by the boiler is absorbed through the tubes.

To raise the pressure of steam to a degree sufficient for the efficient working of a power plant would require either a series of separate boilers or a composite boiler consisting of a number of independent sections, each section forming within itself a boiler, exhaust steam being led to the first, and each subsequent section or boiler being fed by steam from the preceding boiler. In all probability in such a composite boiler the sections in closest proximity to the furnace where the temperature is highest would be capable of producing a pressure difference of from 40 pounds to 50 pounds per square inch.

A diagrammatic view of such a vertical composite boiler is given in Figure 3. The furnace is not indicated, but is intended to be at the bottom, the products of combustion passing up the center of the boiler. Each section

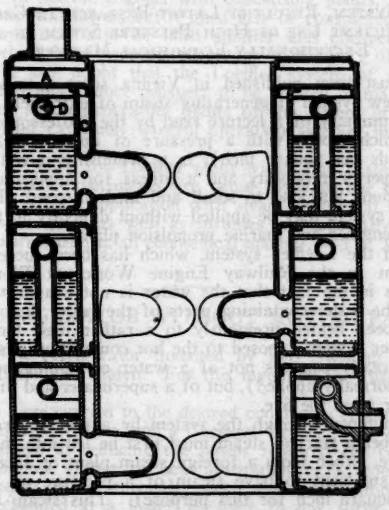


FIG. 3.—REGENERATIVE BOILER.

is divided into two chambers, an outer or main chamber, containing water and steam under pressure, and a feed reservoir. The closed-ended tubes are secured to the walls inclosing the feed reservoir, the closed ends projecting into the flue passage and the open ends communicating with the main chamber. Exhaust steam is fed into the receiving chamber A at the top of the boiler and passes through holes into the feed reservoir C of the first section. Communication between the feed reservoir and the closed ended tubes is through holes in the tubes at the section of low pressure, through which the exhaust steam flows and mingles with the contents of the main

chamber at higher pressure. Steam is conducted away from the main chamber by means of an anti-priming pipe D, and led to a chamber between the first and second sections, and from there through holes to the feed reservoir of the second section.

Should it be found impossible to return the whole of the exhaust steam to the boiler, the water level in the first section must be maintained by means of a pump or injector, or if the exhaust pressure is so low that the pressure in the main chamber of the first section is equal to that of the atmosphere, then the water level may be maintained by direct connection with an external tank, the level of which is kept constant by means of, say, a ball float valve. Each succeeding section would take its feed from the preceding section.

Steam regeneration on the principle roughly outlined is claimed to be a practical proposition and would result in a considerable saving of fuel, thus forming a big step toward cheap power with its consequent beneficial effect toward industry.—"Power," July 21, 1925.

A PROMISING SYSTEM FOR GENERATING STEAM WITH HIGH PRESSURES.

The Loeffler System, Result of Latest Research in Germany, Makes Efficient Use of High Pressure Steam in an Exceptionally Economical Manner.

There have just been published in Vienna some details by Professor Loeffler on his new system of generating steam of extremely high pressures. The paper is a summary of a lecture read by the professor on March 19th. This system, which works with a pressure of up to 100-120 atmosphere (1400-1700 pounds per square inch), is a promising novelty and has the advantage of superior reliability and a fitness for application not only to large power stations but also to small and medium-sized plants. Another merit is that the system may be applied without difficulty to transport craft, such as railway engines and marine propulsion plants.

The novelty in the Loeffler system, which has been successfully applied to a steam plant in the Railway Engine Works at Floridsdorf (near Vienna), consists in the fact that the water is not evaporated by external heat applied to the water-containing parts of the boiler, but by superheated steam that has been heated previously to a rather high temperature in a normal superheater system exposed to the hot combustion gases of the fuel. Thus, the plant really consists not of a water containing boiler and of a superheater (as normally applied), but of a superheater and of an evaporator that is located far from the fire.

The steam is pumped through the system by a special circulating pump. Naturally, when being started, steam must first be led into the system from an external source, either from a foreign steam plant or from an auxiliary boiler, but it is sufficient to have steam of a pressure of 2 atmospheres (28 pounds per square inch for this purpose). This steam is led into the superheater and then into the water-containing evaporating drum, where it begins to generate steam. The pressure is raised gradually up to a working pressure of 100-120 atmospheres (1400-1700 pounds per square inch) in a comparatively short time. In the Floridsdorf plant, foreign steam of 12 atmospheres (170 pounds per square inch) was applied and within an hour the pressure was raised to 100 atmospheres. This is accomplished in a much shorter time than with a normal steam plant of medium pressure.

Speedy starting may be effected, because only the superheater system is exposed to the hot fuel gases. This system consists of tubes of small diameter and is made of Siemens Martin steel. It may be exposed to extreme pressures and temperatures without difficulty, as only pure steam

is contained in it, so that no scale can be formed. The heat transmission is regular, because the steam is moved compulsorily through the system. Tests, which have been made from time to time in the Floridsdorf plant with the superheater material, have been very favorable.

The evaporator is a rather small drum of simple design, which is lighter in weight and much cheaper than normal boilers. Scale formation in it is of no importance whatever, as the heat of the superheated steam is transmitted directly to the water and any scale in the inside of the drum acts practically only as a favorable means of insulation against radiation of heat. The evaporator lies near the steam engine and is automatically regulated by it.

The deciding advantage of the new system lies in the fact that its first cost is by no means higher and is probably even lower than that of a normal steam plant of medium pressure and that its reliability and endurance are superior, so that depreciation may be calculated on a lower percentage. As concerns the economy of generating power, the advantages of steam of extreme pressure are well known. In special cases, the fuel saving may amount to about 50 per cent of the normal rate. A railway engine of 2000 horsepower, in which such results are expected, is now under construction at Floridsdorf. In marine engines with condensing plants, the saving will, of course, be somewhat lower, but it must also be considered that the steam engines themselves may also be built smaller and cheaper for a certain power; a fact which is of especial advantage in the case of marine plants. There is scarcely any doubt that the Loeffler system of generating high pressure steam has much more prospect to be applied to marine plants than the high pressure steam generating plants of the Atmos or of the Benson type, which demand extremely careful operation and are more suitable for stationary plants.

It is intended to work the Loeffler plant with firing systems that may be easily controlled, and experiments are being made to use coal dust firing. As concerns the system's application with steam engines, it has been proved that reciprocating engines as well as turbines are especially fit for making use of high pressure steam in an economical manner.—"The Marine Journal," April 18, 1925.

A REFRIGERATOR WITHOUT MOVING PARTS.

Description of a refrigerator invented by two Swedish engineers, Carl Munters and Baltzar von Platen. For this invention the 1924 Polhem medal was awarded.

The inventors have carried to the desired conclusion researches and inventions by a Frenchman, Carres, and a German, Geppert.

The former aimed at reducing the power required for compression purposes in the ammonia system of heat absorption (*i. e.*, cooling). He still retained a reducing valve, through which the liquid ammonia at higher pressure was expanded at a lower pressure to gas form, thereby seizing upon any available heat and causing the desired refrigerating effect.

Geppert carried the Frenchman's idea a step further and discovered that refrigeration could be obtained without any difference of pressure if the apparatus was filled with a neutral non-absorbent gas, which, by its presence, bridged the pressure gap between the evaporator on the one side and the generator and absorber on the other. Thus Geppert eliminated the compressor and reducing valve but the absorbing liquid (water) required mechanical assistance for its circulation, and a further source of inefficiency inherent in Geppert's design, in which the generator and absorber were combined in one apparatus, necessitated for its partial elimination the introduction of a mechanically driven fan.

The Munters-Platen system is a self-contained apparatus in which the same pressure exists in all parts, and in which heat applied to one portion of an enclosed system produces cold in another portion of the system.

The following is the method adopted, but it should be pointed out that calculations have been made for any combination where:

A is any suitable gasifying medium—in the following description it is ammonia.

B is any absorbing medium—in the following description it is water; and

C is any non-absorbent gas (whether capable or not of being condensed)—in the following description it is hydrogen.

The Munters-Platen apparatus consists of:

An evaporator K, in which in this instance a strong solution of ammonia is heated and gives off ammonia gas to—

A condenser X, in which the gas is cooled and liquefied;

A generator G, in which the liquid ammonia is introduced and gasified by the absorption of heat from its surroundings. G is therefore the cooling body;

An absorber A, where the ammonia gas is absorbed by water (and incidentally heat given off in the process); and

A temperature exchanger T, whose function will be described.

The automatic circulation of ammonia gas and absorbing water is brought about by very ingenious means.

The evaporator K contains a strong solution of ammonia in water. At its base there is a heating body which is heated either by lamp or, better still, by an electric resistance. Round this body is coiled a pipe which projects above the surface of the liquid in the evaporator. When heat is applied, this pipe becomes so hot that the water inside it boils, and rising, discharges into the evaporator, and ammonia gas is driven off. By this means ammonia water is continuously drawn from the absorber A through the inner coil of the temperature exchanger into the evaporator K, and, as stated, ammonia vapor is driven off from it.

We shall now follow the ammonia-gas circulation, and later return to follow the adventures of the water.

Proceeding from evaporator K the ammonia gas passes through the condenser X, where it becomes liquid and proceeds to the generator (the refrigerating portion), which is filled with hydrogen.

In the generator the liquid ammonia, by cooling its surroundings, becomes gasified and mixes with the hydrogen. The resultant gas mixture is heavier than is the pure hydrogen, and therefore the mixture, by reason of its weight, passes through pipe R″, into the absorber, where it meets a shower of water. The water absorbs the ammonia gas, which gives off its latent heat (and cooling of this portion is therefore necessary—it is cooled by water, which later proceeds to the condenser X), and the hydrogen, denuded of its heavier ammonia-gas content, becomes lighter than was the gas mixture, and so rises and passes again into the generator by way of pipe R′, and so continues to circulate automatically.

A feature introduced into the actual apparatus as made, but which for the sake of simplicity, is not shown on this diagram, is that pipes R′ and R″ are laid one inside the other, forming another temperature exchanger, in order that the cold gas mixture on its way to the absorber may become heated and the warmer hydrogen on its way to the generator may be cooled by the exchange of heat on the contraflow system.

Now, we must return to the question of the circulation of the absorbing agent (water).

The water partly denuded of its ammonia content on entry into the upper part of the evaporator K, is still further denuded of ammonia by the heat applied at the evaporator base. This water flows away by gravity from the evaporator by means of a pipe situated through the outer coil of the tem-

perature exchanger T, being cooled as it proceeds, and enters the absorber A at its upper end. Being relatively free of ammonia it readily absorbs the ammonia gas contained in the ammonia and hydrogen atmosphere existing in A and falls to the bottom of A, being once more a strong solution of ammonia. It is withdrawn from A through the temperature exchanger, being heated as it passes along until once more it passes through the heating coil in the evaporator already referred to, and so, being heated violently, rises and discharges once more into the evaporator. This completes the description of the cycles for the gasifying medium, the absorbing medium, and the non-absorbent gas medium, which are continuous.

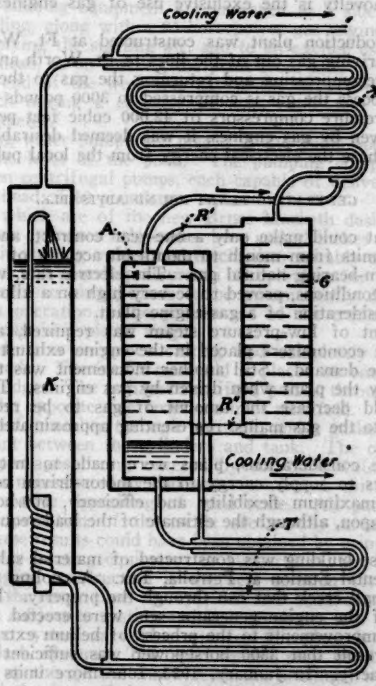


DIAGRAM OF THE MUNTERS-PLATEN REFRIGERATOR.

It will thus be seen that the only motive force for the whole apparatus which sets these cycles in operation is derived from a heating coil, and that by the application of heat to this one portion the refrigeration desired is obtained in another portion and, as already stated, the pressure in the apparatus is the same in all parts and the whole forms a hermetically sealed system. (R. W. in "The Machinery Market," No. 1278, May 1, 1925, p. 21, 3 Figs., *d.*)—"Mechanical Engineering," July, 1925.

GENERATING UNITS OF UNITED STATES HELIUM GAS POWER PLANT.

By Z. W. Wicks, Lieutenant-Commander, U. S. Navy.

Helium has removed most of the danger attending the use of dirigible air ships through its replacement of the highly explosive hydrogen gas. Only a few years ago helium was considered a gas existing only in the sun. Consequently, the discovery in the natural gas from the Petrolia, Texas, fields of this non-explosive gas gave the United States Government a decided advantage in aviation. While the process of recovery is interesting, to engineers the chief novelty is the exclusive use of gas engines in the helium recovery plant.

The helium production plant was constructed at Ft. Worth, Texas, in 1920 and 1921, deriving gas out of the lines to Ft. Worth and after extracting the helium, recompressing and returning the gas to the pipe lines. In the extraction process the gas is compressed to 3000 pounds per square inch and since high-pressure compressors of 42,000 cubic feet per hour capacity had not been driven by gas engines, it was deemed desirable to use motor drive and to purchase the electrical energy from the local public utility.

GENERATING PLANT FOUND ADVISABLE.

The government could make only a one-year contract, and the load fluctuated on wide limits from month to month on account of the variation in the flow of helium-bearing natural gas. The electric rate, while reasonable in view of these conditions, proved to be very high on a kilowatt-hour basis. This led to a consideration of a gas-engine plant.

A small amount of low-pressure steam was required, and calculations indicated that the economizers placed in the engine exhaust lines would be able to supply the demand. Still another inducement was the reduction in power required by the plant when driven by gas engines. The gas used by the engines would decrease the amount of gas to be recompressed and delivered back into the gas mains, representing approximately one-eighth of the gas processed.

Based on these considerations, plans were made to install gas engines driving alternators to supply energy to the motor-driven compressors. In order to obtain maximum flexibility and efficiency, 500-horsepower units were determined upon, although the estimate of the load requirements totaled 8000 horsepower.

The power-house building was constructed of material salvaged from the Helium Experimental Station at Petrolia, Tex. A cooling pond was made by damming a small creek that ran through the property. In March, 1924, the first three of the engine generator sets were erected. By December many important improvements to the process of helium extraction had been made, with the result that 3500 horsepower was sufficient to operate the plant at full capacity. In January, 1925, four more units were installed, completing the seven necessary.

Each of the gas-engine units consists of two Bruce-Macbeth four-cylinder 250-horsepower engines with a 435-kva. 2300-volt three-phase 60-cycle alternator placed between the two engines and connected to the shafts by rigid couplings. With a flexible coupling there would be experienced a backlash from one engine to the other, which would probably result in a broken crankshaft in a short time. With the rigid coupling the two machines become, in fact, an eight-in-line engine with the generator interposed between the fourth and fifth cylinders. No difficulty has been experienced in keeping the bearings lined up. Each main bearing has its own alignment pin, and

one gage pin fits all bearings so that they can all be brought to the same height. The generator bearings are set with a tram furnished by the manufacturer of the engines, and after a little experience a good mechanic has no difficulty in getting almost perfect alignment.

ENGINE AUXILIARIES.

Each engine is fitted with an exhaust economizer, or boiler, which under test has evaporated 2.5 pounds of water from 100 degrees F. to steam at 15 pounds gauge per engine brake horsepower output.

The engine-generator units are arranged in two banks, three in the south and four in the north bank. Two 150-kw. motor-generator exciter units with automatic starting equipment are arranged in a side bay on the east side of the building, along with a 25-horsepower gas engine belted to an air compressor supplying the starting air, and to an exciter used only when the main exciter is dead, as on starting the plant from idle.

In the north end of the west side bay is the battery of 14 dry positive gas meters, each having a capacity of 5000 cubic feet per hour, and a recording calorimeter.

The circulating-water pumps are placed in the pump-house built adjacent to and below the level of the pond. The pumping equipment consists of two motor-driven centrifugal pumps, each capable of delivering 2000 gallons per minute at a head of 50 pounds per square inch.

The engines, which are of the new Bruce-Macbeth design known as the "Tank" type, have cylinders 14½ inches bore and 17 inches stroke. They are equipped with air-starting valves and mechanism suitably arranged to admit compressed air at 350 pounds pressure to all cylinders for starting purposes. This mechanism is so arranged as to be automatically disconnected after the engine is in operation. The pistons are of the trunk pattern and are fitted with five cast-iron piston rings.

The open tank is a one-piece casting and completely incloses all four cylinders and heads. The intake manifold is placed on one side of the tank near the upper edge. The exhaust manifold passes through the tank near the bottom on the opposite side from the inlet manifold. The cylinders are bolted to the crank chamber through the tank, and a gasket is used to make a watertight joint between the cylinders and tank. The cylinder heads are without water passages, being covered by the tank water. This type of construction was necessitated by bad water conditions existing at this plant, which cause excessive depositing of hard scale. All parts of the engine subject to the formation of this scale are readily accessible and easily cleaned. The same results could have been obtained by using a closed cooling system with water-treating equipment or evaporators, but it is believed that this type of engine construction is the best solution for the local water conditions. At the same time the cylinder and head construction is much simpler and less subject to casting and heat strains.

The lubrication system as worked out for this plant is a modification of the original splash system. The solid splash pins fitted to the crankpin bearing to pick up the oil and throw it on the bearings, were replaced with hollow pins and the lower halves of the bearings were drilled through in the wake of the pins. By using a large plate-glass door in the crank chamber, the best shape and dip of the pins were determined so as to give the maximum lubrication to the crankpins and ample to the cylinders with the minimum agitation of the oil in the crank chamber. Each chamber is fitted with an adjustable overflow so that a fixed oil level is maintained in all compartments, all overflows being piped to an underground tank. From this tank a suction is taken by one of two Viking spur-gear pumps, and the oil is discharged to the oiling system through an oil cooler at a pressure of about 20 pounds per square inch.

Each engine is fitted with four sight-feed oilers into which the oil is discharged and flows directly to the caps of the main bearings. In addition oil thrown by the splash pin is caught in a pan near the top of the crank chamber and flows to the main bearings. The oil in the underground tank is cleaned for about six hours daily by means of a centrifugal purifier, and all new oil is introduced into the system by dumping it into this same tank. Additional cylinder lubrication is obtained by forced-feed lubricators eccentric driven from the camshaft and so timed as to deliver the oil into the space between the first and second piston rings.

Only one mixing valve is used for both engines of a unit, and it is located centrally between them. This assures that all eight cylinders will get the same mixture and each engine will perform its proper share of the work put on the unit. The governor is of the flyball throttling type and controls the mixing valve. While the action of this governor is quite satisfactory, it is believed that more sensitive control of speed and, consequently, of voltage could be obtained if the work on the governor was restricted to the operation of a pilot valve, which in turn would control an oil cylinder actuating the mixing valve.

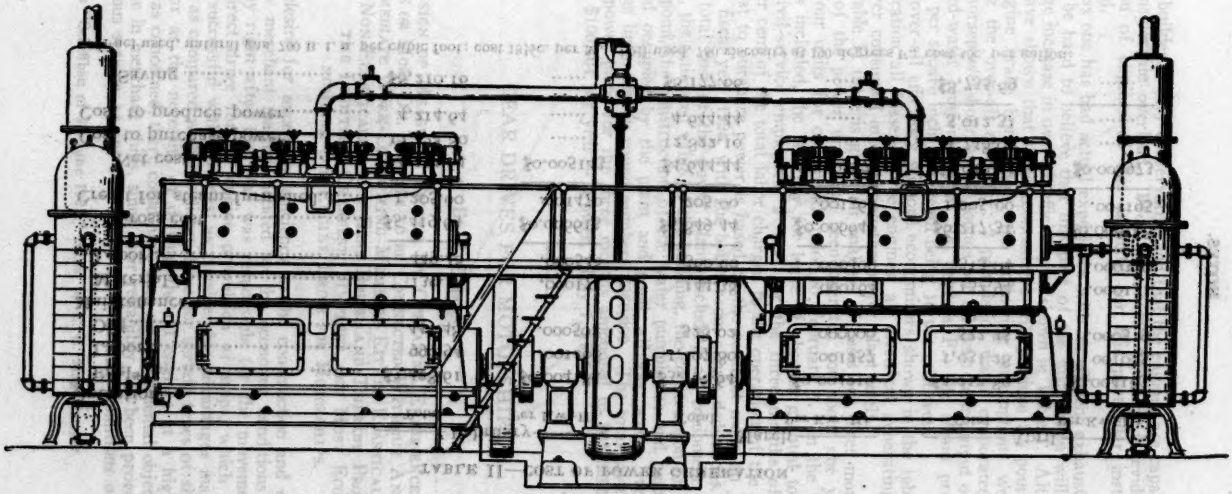
The panels and switchboards are arranged facing the power units and all rheostats are grouped behind the board. As the panels are set up on the floor level, the view of the synchroscope is blocked from several of the units. In order to facilitate synchronizing the units, two additional synchroscopes have been installed, back to back, well overhead and between the two tanks of units so that one of the instruments is visible from any of the mixing valves. In order to encourage better adjustments of the mixing valves and to cut down the time necessary for the enginemen to go from one mixing valve to another, the operating platforms of all engines have been connected by a walkway. The small cost of this walkway has been more than paid for by the better care given the machines.

The operating data for three months are shown in Table I. The rated capacity per unit, including the conversion factor of horsepower to kilowatt-hour, the alternator efficiency and the necessary power to operate the motor-

TABLE I—OPERATING RECORD OF NATURAL GAS ENGINES.

Date	February	March	April
Number of units operating	3, 4, 5 & 6	3, 4, 5	3, 4, 5 & 6
Unit-hours operating	2645	2785	3062
Maximum demand, kilowatt-hours	1775	1500	2000
Minimum demand, kilowatt-hours	300	740	800
Average demand, kilowatt-hours	1220	1184	1400
Total output, kilowatt-hours	819,500	881,000	1,008,500
Average load on each, kilowatt-hours	309.8	316.3	309.8
Unit load factor, per cent	99.14	101.22	99.14
Plant load factor, per cent	55.75	54.13	64.03
Total gas consumed, thousand cubic feet	18,533	20,047	22,800
B.t.u. per cubic foot of gas	700	700	700
Consumption of B.t.u. per kilowatt-hour	15,830	15,928	15,825
Over-all efficiency without economizers	21.57	21.44	21.58
Over-all efficiency including economizers	41.78	41.53	41.80

generator exciter unit, the oil pumps, the oil purifier and power-plant lights, is 312.5 net kilowatts delivered at the switchboard. The total output as given in the table is the power consumed in the helium plant outside the power house. It will be observed that the unit load factor is practically unity for the three months. In order that the average load per unit for the month can be that high, it is clear that the rated capacity as given by the



ARRANGEMENT OF TWO ENGINES DIRECT CONNECTED TO A SINGLE GENERATOR.

TABLE II—COST OF POWER GENERATION.

	February		March		April	
	Total	Per Kw.-Hr.	Total	Per Kw.-Hr.	Total	Per Kw.-Hr.
Operation						
Fuel*	$3,428.61	$0.004184	$3,708.64	$0.004210	$4,217.80	$0.004183
Labor	990.94	.001208	1,107.80	.001257	1,031.28	.001025
Oil†	441.48	.000502	528.02	.000600	522.25	.000519
Maintenance						
Material	139.13	.000171	144.38	.000164	132.94	.000131
Labor	449.48	.000548	360.60	.000409	313.04	.000310
Gross cost	$5,419.64	$0.006613	$5,849.44	$0.006640	$6,217.31	$0.006168
Credit for steam furnished	1,205.00	.001470	1,205.00	.001368	1,205.00	.001195
Net cost	$4,214.64	$0.005143	$4,644.44	$0.005272	$5,012.31	$0.004973
Cost to purchase power	12,424.80	.....	12,822.10	.....	13,748.00	.....
Cost to produce power	4,214.64	.....	4,644.44	.....	5,012.31	.....
Saving	$8,210.16	.....	$8,177.66	.....	$8,735.69	.....

*Fuel used, natural gas, 700 B. t. u. per cubic foot; cost 18½c. per M. †Oil used, 750 viscosity at 100 degrees F.; cost 45c. per gallon.

NOTES.

engine builder is very conservative and that the engines are capable of carrying considerable overload. From this same figure it is evident that the selection of engines of this capacity for this particular installation was justified.

Unless one has had actual experience with an efficient exhaust economizer, it will be hard to believe that this part of the installation will bring about the large increase in over-all efficiency as shown in Table I. Although actual tests have shown that these economizers will produce 2.5 pounds of steam per engine brake horsepower, the following assumptions were made in figuring the over-all efficiency: 2 pounds of water evaporated per engine brake horsepower-hour or 3.2 pounds per kilowatt-hour output of plant, with the feed-water temperature at 100 degrees F.; the steam pressure at 15 pounds per square inch, the steam being 10 per cent wet.

The over-all efficiency without economizers shown in the table takes into consideration all losses in the engines, all electrical generating losses and all power consumed in operating the power plant itself.

In Table II there is given an analysis of costs for three months in 1925. Three of the power units had been operated for about one year and the other four units for one month prior to February 1. In the labor charge there is included, the full pay for leave of absence granted to all government civil-service employees, a total for the three months of $477.58, or 11¼ per cent of the total labor charged to the operation of the power plant. The cost to purchase power is based on the contract made on July 1, 1924, for the fiscal year 1926, with the boiler-room clause deductions.

The total cost of the power plant, including all equipment and erection thereof, the estimated value of the building, the construction of the 1,500,000 gallon pond, the construction of the water pumphouse and pumps, value of the land occupied by the plant and pond, all gas, water, steam, air and sewerage lines, was in round numbers $360,000. The total saving applicable to the power plant for the three months amounted to $25,123.51, or at the rate of $100,404 per year.—"Power," July 7, 1925.

GEAR DRIVES FOR MOTORSHIPS.*

By R. Dreves.

Discussion of Measures Proven By Operating Experience to Enable One or More Oil Engines to be Connected Without Any Intermediate Work-Consuming Members Either Elastically or Non-Elastically to a Gear Drive, and Thereby Produce the Lightest, Cheapest, Simplest and Most Economical Oil Engine Drive for Motorships.

The desire for maximum efficiency in the construction and operation of the new merchant marine dictated by the economic conditions existing in Germany right after the War was responsible for the movement to adopt the indirect drive for oil engine propulsion of ships, which has already proved successful with turbines, as considerable advantages may be gained thereby as compared with the direct drive for certain types of ships.

As far as the necessity of driving the oil engines at a higher rotative speed was concerned, there could have been no fundamental objection against their use in merchant ships, since their reliability has been proven, if not in continuous service, at any rate under most difficult conditions of submarine

* From "Schiffbau of 24 June 1925," trans. by E. C. Magdeburger.

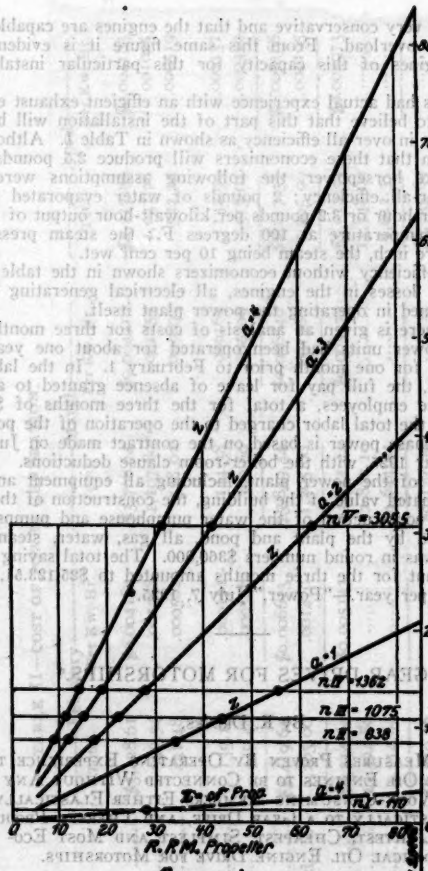


FIG. 1.—EXCITING FREQUENCIES AND NATURAL FREQUENCIES OF A TURBINE GEAR DRIVE WITH SINGLE REDUCTION AND ELASTIC CONNECTION BETWEEN TURBINE AND PINION.

z = frequency of exciting force.

nI-nV = natural frequency of mass elastic system.

• = critical speeds.

a = number of impulses per revolution of pinion.

service. To suit the requirements of merchant marine for relatively higher reliability and longer life, both engine speed and mean effective pressure have been suitably reduced.

The intermediate gear requirements, however, could not be so easily clarified. Electric drive and the use of hydraulic gear could not be considered on account of their reduced efficiency and serious objections had to be overcome before the highly efficient gear drive could be used. It was noted that all writers at home and abroad, who have discussed the subject of indirect drive for motorships, considered the gear drive as unsuitable because the torque of an oil engine varies considerably and rather suddenly and the gear transmission could hardly be expected to be reliable.

Since, in the meantime, the first motorships *Havelland* and *Munsterland* (with recently added *Vogtland*) equipped by Blohm & Voss with oil engines driving through mechanically coupled gear transmissions have demonstrated their absolute reliability in several years of service, including most difficult maneuvering and extraordinary sea conditions (Taifun), it may especially be interesting to analyze the important reasons that bridged the gap between the once prevailing negative expert opinion and the success actually achieved.

Before the above-named motorships were built, gear drives were tried out only with high speed turbines and the experiences with such installations notably abroad were not always quite satisfactory, the gear teeth would suffer damage after comparatively short periods of service. It would appear from these occurrences, the causes of which were not satisfactorily explained, that if such consequences would show up in turbine drives with their perfectly even turning moment, it surely would be impossible to make such a sensitive gear transmission reliable under the pulsating torque of an oil engine, even if an attempt were made to reduce pulsations by fitting heavy flywheels.

Actually the conditions are quite different. A critical comparison of both installations permits one to see the cause of possible danger and the means for its elimination, if as the basis of the ideal conditions is established the well known requirement, that gear transmissions must be subjected only to positive turning moments throughout the range of their operation.

Investigations on the suffering turbine installations† established, as the cause of the above mentioned damage to gear teeth, minute inaccuracies in the manufacture of these gears. These inaccuracies would produce pulsating forces in the gears, which periodic forces, when synchronized with the natural period of the mass-elastic system, would result in forces many times larger than the original forces and cause damage and even breakage of gear teeth.‡

The explanation of this effect becomes more clear when not only the force alone is considered, but also the work done by the force on the whole system and the dissipation of this work. The forces themselves can, for this purpose, be regarded as harmonious functions, *i. e.*, varying in accordance with the sinus curve. Then it becomes clear that the effect produced by the resonance or synchronism of the exciting force depends not only on its amplitude, but also on its velocity and the magnitude and velocity of the work consuming resistance. The velocity of the exciting force is by constant amplitude a function of the frequency of its recurrence in a unit of time, or periodicity. This frequency is of particular importance because it serves to measure the "hammering effect" of the force. A given force re-occurring more frequently—with a higher frequency—has more of a hammering effect, hence is more dangerous than the same force but of a lower frequency. Accordingly one can differentiate between a hard and a soft exciting force.

† Dr. Frahm—"Gear Drives for Turbines and Motorships," paper read before Schiffbautechnische-Gesellschaft, 1924.

‡ Gümbel-Torsional vibration in gear drives for ship propulsion—VDI, 26 Aug. 1922.

If practical limits do not permit further reduction of magnitude and frequency of the exciting force there remains in some cases still the possibility of wiping out the work of this force by robbing it of its leverage, *i. e.*, by placing its point of application—the gear transmission—in the node of vibration.* This is only possible, however, for one mode of vibration. It is, therefore, always desirable to be able to influence the conditions of the work of dissipation (damping). In order to be able to dissipate the damping work, which is equal to the work of the exciting forces, without any harm to the gear transmission, especially in cases of forces of high frequency, no large masses should be direct (hard) coupled with the pinion, because due to high frequency, even with very small amplitudes of thousands of an inch, the large accelerations and retardations coupled with the large masses produce excessive tooth pressures. In fact the damping forces must be provided with larger amplitudes which can be done by inserting between the masses of the gear transmission and those next to them sufficiently elastic connections. One of such type of "soft couplings" of masses accomplishes the necessary spreading of damping work and reduces the magnitude of the damping forces in the gear transmission to perfectly safe figures.

Briefly stated the damage to the gear teeth of turbine gear drives was due to hard excitation and hard coupling of masses. It was overcome by (*a*) placing the unavoidable synchronous critical speeds outside of the working speed range; (*b*) reducing the hard excitation through improvements in gear cutting methods to the practically possible minimum; (*c*) avoiding hard connection between the gear drive masses and the rest of the masses.

Figure 1 shows the frequencies of the exciting force and of the free vibration of the system for a turbine gear drive, with the critical speeds at the intersection of these two sets of lines. These, however, by using the above named remedies become meaningless and the full speed of the vessel is quite free of them.

In oil engine gear drives the exciting forces in the drive itself, caused by similar reasons as described above, are of small importance, due to their low frequencies. If frequency, *i. e.*, the number of vibrations per minute or V.P.M. be designated by Z, the number of impulses per revolution by *a* and the number of revolutions per minute or R.P.M. by *n*, then $Z = a \times n$. The same number of impulses per revolution of the pinion as has been established for turbine pinions can properly be assumed for oil engine driven pinions and the frequency of impulses is then proportional to the ratio of speed between oil engine and turbine which can be assumed as about $250 \div 2500 = 1 \div 10$. Hence, the accelerations of the motion produced by these impulses are only 1/100 of those in the turbine gear drive and consequently these exciting forces are of no practical importance in the oil engine gear drive.

By far more important, on the other hand, are the periodic turning forces of the oil engine itself. Whereas, the exciting forces originating in the gear drive cannot be calculated in advance as to their number and magnitude because they are functions of accidental circumstances in manufacturing, the exciting forces of the oil engine are well enough known as to number, magnitude and phase to be able to predict their influence on the system by analysis. Whereas, the influence of the exciting forces of the gear drive is rather difficult to correct because they act in the same place where they are generated, the forces transmitted from the engine to the gear drive, due to their separation from the gear drive, can be more easily made harmless by insertion of suitable damping between the two.

Aside from the conditions of synchronism, which will be discussed later, the simplest, cheapest and most reliable means for weakening the pulsations of the turning forces is the flywheel. The turning effort of the crankshaft, which in many types of oil engines varies many times during one revolution

* Gümbel-Torsional vibration in gear drives for ship propulsion—VDI, 26 Aug. 1922.

from positive to negative, can be so evened out by the flywheel that the accelerations and retardations beyond it will result in only minor variations from the positive mean tooth pressure.

The calculation of accelerations on the pinion pitch circle for multi-cylinder engines can be made with sufficient accuracy by means of the basic harmonic turning force of the engine type without considering the overtones or minor harmonics of the turning effort diagram. The tooth pressure necessary to accelerate the mass of the large gear wheel

$$p = \frac{P_h}{m}$$

where p = acceleration or retardation on the pitch circle of the gear transmission

P_h = basic harmonic turning force of the engine, referred to the pitch circle of the gear drive,

m = mass of the flywheel, referred to the pitch circle of the gear drive.

By using mean values the following formula may be developed for all engine types with equidistant firing of cylinders

$$P_h = 145{,}000 \frac{IHP \times p_h}{n \times d}$$

where

d = pinion diameter in centimeters.

p_h = basic harmonic turning force of a single cylinder single acting four cycle engine in kilograms per square centimeter of piston area.

Figure 2 shows all the harmonics of a single cylinder, single acting four cycle engine in kilograms per square centimeter, wherefrom the basic harmonic p_h of any engine type can be taken. The basic harmonic is such harmonic turning force, whose number of impulses per revolution (order number A), A = 360 ÷ angle between explosions, so that for single acting six cylinder four cycle engine the basic harmonic $A = \frac{360}{120} = 3$ order, and from Figure 2, p_h = 1.7 kilograms per square centimeter, whereas, for a single acting six cylinder, two cycle engine, as well as for a double acting six cylinder two cycle engine $A = \frac{360}{60} = 6$, and p_h = .39 kilograms per square centimeter, except, for the latter, two impulses fall together and p_h = .78.

The problem to subject the gear drive only to positive turning moments at speeds outside of synchronous or critical speeds is very simple, however, special corrective measures are necessary to prevent any damage to gears from critical speeds.

Critical speeds occur when the frequency of the periodic (harmonic) forces acting on the crankshaft is equal to the frequency of the natural or free vibration of the system consisting of the rotating masses of the engine, flywheel, gear drive and propeller, or in other words, when synchronism or resonance occurs.

The first problem of the designer is, therefore, not to permit such synchronism or equality between the frequency of the exciting forces and the natural period or, if that is impossible, to place the critical speed outside of the operating range of engine speeds. It becomes necessary, therefore, to analyze the system with a given set of dimensions for cylinder bore, crank and line shaft diameters, as well as given masses of flywheel and gear drive, for torsional vibration which necessitates knowledge of modes of vibration and forms of excitation.

MODES OF VIBRATION.

Since four main masses make up the system, three different modes of vibration are possible. These are shown on Figures 3-10, under A when between flywheel and pinion an elastic connection is made, and under B when a practically non-elastic connection exists.

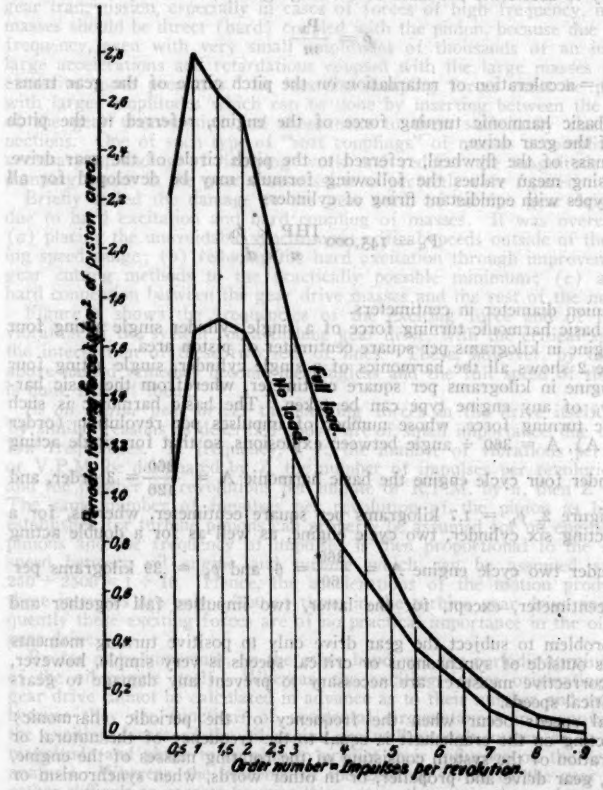


FIG. 2.—PERIODIC TURNING FORCES OF A SINGLE 4-CYCLE CYLINDER.

The first mode of vibration has one node with the two vibrating masses consisting of engine and flywheel against the propeller, whereas the gear drive masses vibrate either with the engine or with the propeller. In case the gear drive is located in the node, it does not participate in the vibration, however, such an arrangement with the excitation originating in the engine or propeller is not successful.

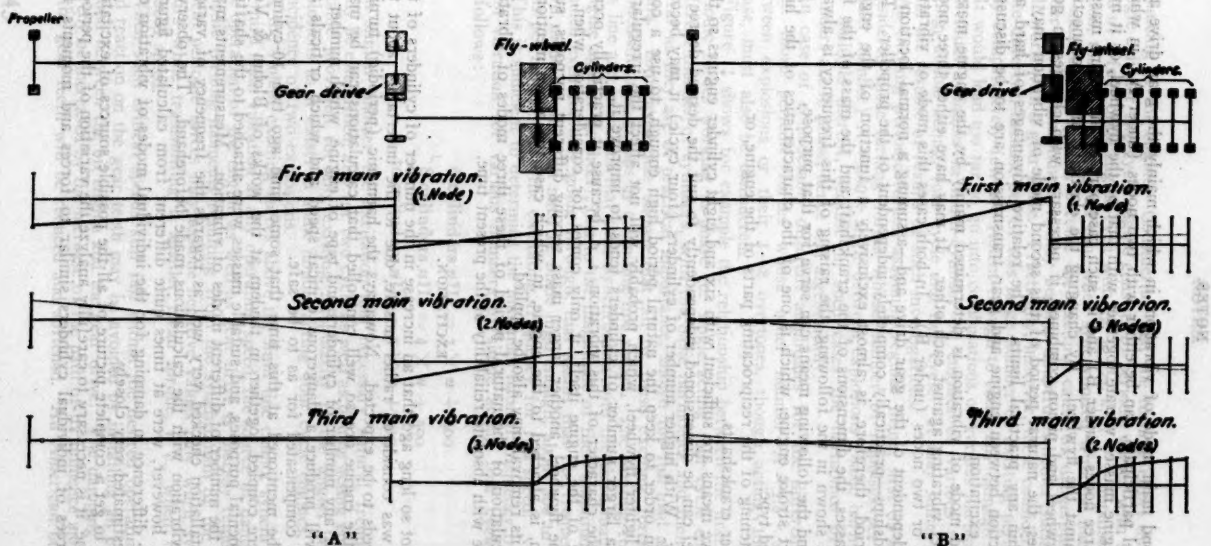


FIG. 3-6.—WITH ELASTIC CONNECTION. FIG. 7-10.—WITH HARD CONNECTION.

FIG. 3-10.—MODES OF VIBRATION OF OIL ENGINE GEAR DRIVE INSTALLATIONS.

The second main mode of vibration, in which mainly the gear drive and the flywheel participate, can be either with two nodes (under A), in which case the engine masses vibrate together with that of the flywheel, or it may build up three nodes (under B) and in such case part of the engine masses vibrate against the flywheel. By changing the elasticity of the connection between flywheel and pinion combined, if necessary, with changes in gear drive masses, the natural period of this second mode of vibration may be varied within any practical limits. The relative advantages of hard and soft connection between engine and gear transmission are to be discussed later under excitation.

The third mode of vibration is determined mainly by the engine masses and flywheel vibrating against each other. It may have either three nodes (under A) or two nodes (under B). In both cases this mode of vibration is quite independent of the gear drive and—assuming a normal location of engine amidships—practically completely independent of the propeller. The natural period, therefore, is almost exclusively a function of the engine rotating masses, the dimensions of the crankshaft and the mass of the flywheel. As shown in the following, the raising of this frequency is always desirable and the following means can serve for that purpose.

(*a*) short stroke engine, which is one of the characteristics of the high speed type.

(*b*) lightening of the reciprocating parts of the engine, or

(*c*) larger crankshafts.

The above means are sufficient with six and eight cylinder engines so that the flywheel can be dimensioned independently to suit the desired degree of irregularity. With higher number of cylinders (four cycle) it may become desirable, in order to keep the natural period high enough, to use a comparatively lighter flywheel, which probably will not affect the irregularity either, as a larger number of cylinders tends to improve it.

Due to the character of this vibration, *i. e.*, because it practically occurs only inside of the engine itself, it only comes for consideration when, in place of the flywheel, another larger mass serving different purposes, such as a clutch, is attached to the engine, in which case the above mentioned means for its removal may also be applied.

The calculation of the natural period of these three modes of vibration can be done with absolute reliability at the present time.

EXCITATION.

It was not so long ago that an increase in the number of cylinders of the oil engine was considered rather unsafe, due to the uncertainty about the critical speeds to be expected. Nowadays the harmonic (periodic) turning forces of the engine are so well controlled that calculations can be made to show for any number of cylinders and type of engine, which number of impulses will produce a dangerous critical speed and which criticals are sufficiently compensated for as to be safe.

It may be mentioned at this time that some time ago two 10-cylinder engines were coupled together in tandem at the works of Blohm & Voss for experimental purposes, and suitable masses were attached to the shafting to increase the number of different modes of vibration. Measurements made on this installation checked very well as regards the frequency of various modes of vibration with the calculations made beforehand. The observed amplitudes, however, were at times quite different from calculated figures, because the difference in damping for the individual modes of vibration can hardly be estimated very closely.

In order to get a complete picture of all the possible sources of excitation in the engine, it is necessary to carefully analyze the variation of the periodic turning forces of individual cylinders similar to forces and moments for

balancing of the engine. Considering that the turning effort diagram of a single cylinder two cycle engine consists of periodic forces with 1, 2, 3. . . . and of a four cycle engine even with ½, 1, 1½, 2, . . . impulses per revolution, that theoretically all can, when synchronized with the natural period of mass-elastic system, become the exciting cause of dangerous vibration, under such circumstances it may become doubtful whether such an engine will work with a gear drive. Practically, however, these considerations look quite different, because out of this large number of exciting periodic forces only comparatively few are violent enough to cause any damage.

It would lead too far to go into combinations of various harmonics of the individual cylinders in multi-cylinder oil engines of various types—single acting, double acting, four cycle and two cycle. It may suffice to keep in mind the resumé of results of such investigations.

As exciting periodic forces, assuming regular firing, first of all must be considered those turning forces whose number of impulses per revolution (order number) $A = 360 \div$ angle between explosions (basic harmonics) and the multiple thereof, because in such cases the harmonics of individual cylinders are in phase, hence add algebraically.

In case of considerably different amplitudes for the different cylinders even those periodic forces whose order number $A = 1.5 \times 360 \div$ angle between explosions or their 1—2 overtones, though acting 180° out of phase, must also be considered.

Figure 11 shows for example the relationship between the first mentioned periodic forces working in phase for single acting four cycle engines with different number of cylinders, with the basic harmonics and their overtones as far as they are of practical importance, shown for full load and no load limits. Experience has shown that all of these exciting forces may be divided into two groups by a limiting line shown. The exciting forces above this line are so violent that their synchronism with the third main vibration (engine against flywheel), due to the magnitude of their amplitude, must be avoided even for only going through, that means that the frequency of the third mode of vibration must be made high enough, about 300 to 400 vibrations per minute above the excitating frequency for full speed.

This is sufficient to determine the minimum natural frequency of the third main vibration for various numbers of single acting four cycle cylinders as follows:

$$6 \text{ cylinders } n\text{III} = 3 \times n + 300$$

$$8 \text{ cylinders } n\text{III} = 4 \times n + 300$$

$$10 \text{ cylinders } n\text{III} = 5 \times n + 300$$

In similar manner the excitation and necessary natural frequencies can easily be established, also for two cycle as well as double acting engines, with suitable provision for any existing irregularity in the angles between the explosions as it exists in some double acting four cycle engines, as well as coincidence of two explosions.

If the violent exciting forces above the limit line cannot synchronize with the third main vibration at any operating speed, the source of danger for the safe operation of the engine as well as gear drive is thereby removed. The periodic turning moments resulting from the synchronism of the other less violent periodic forces may assume negative values only when synchronism occurs while transmitting a smaller mean torque, *i. e.*, when running at reduced speed.

When such critical speeds still fall within the desired operating range a hard reaction on the gear teeth may be avoided by installing an elastic connection between the oil engine and the gear drive. The damping effect will thereby be increased by the increase of the path of its influence and reactions on the gear teeth will be reduced and softened. An American builder uses

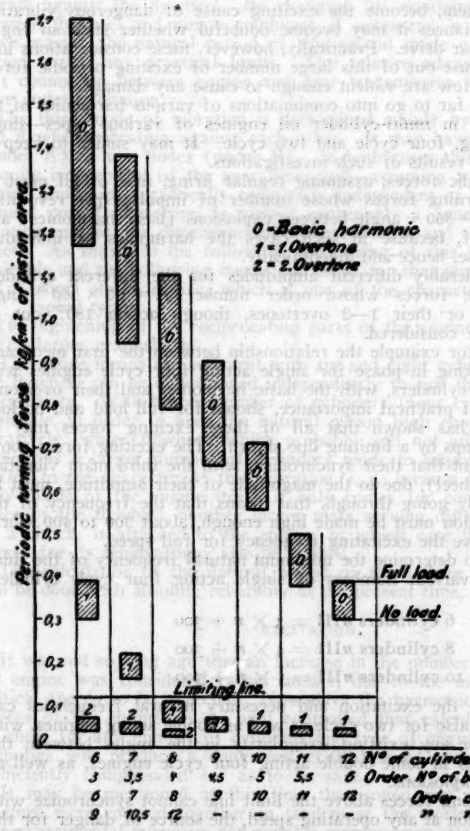

Periodic turning force kg/cm² of piston area
0 = Basic harmonic
1 = 1. Overtone
2 = 2. Overtone
Full load.
No load.
Limiting line.
6 7 8 9 10 11 12 N° of cylinders.
3 3,5 4 4,5 5 5,5 6 Order N° of basic harmonics.
6 7 8 9 10 11 12 Order of 1. Overtones
9 10,5 12 - - - - " " 2. "

FIG. 11.—BASIC HARMONIC EXCITING FORCES AND THEIR OVERTONES FOR SINGLE-ACTING, 4-CYCLE ENGINES.

a special construction of spring coupling to provide the elastic connection, but from the point of strength a solid or a hollow shaft is the best and safest torsional spring.

The installation of such an elastic connection gives to the second main vibration (engine and flywheel against the gear drive) a natural frequency that is lower than the frequency of the third main vibration and places it into the operating range of the engine (Figure 12). It must, therefore, be placed so low that it will be passed through by the exciting frequency of the basic harmonic at as slow speed as possible. This determines the maximum natural frequency of the second main vibration and, if it be required for instance that the critical speed of the basic harmonic should be at ⅓ of the full speed, then n_{II} equals $\frac{n}{3}$ times 360 divided by the angle between explosions. This determines also the necessary elasticity of the connecting shaft.

If it is possible to keep the whole operating range from slow to full speed free from critical speeds that are excited by periodic forces above the limiting line for the third main vibration and to avoid negative moments and to place the remaining less violent critical speeds sufficiently far away from any operating speed, then the elastic connection between the engine and the pinion can be eliminated.

Finally, the first main vibration (engine against propeller) must also be considered. The natural period of this vibration with engine amidships is about $n_I = 120$ to 130 V.P.M. and only very little higher than in turbine installations. It follows, therefore, that even for a six cylinder, four cycle engine with lowest and most violent basic harmonic, the critical speed caused by it shows at about 15 R.P.M. of propeller and is only passed through when starting up, hence is of no practical importance. Any possible excitation through the propeller manifests itself no different from the turbine installations.

The above mentioned periodic forces that are 180 degrees out of phase can be determined only by the aid of the deformed elastic line. They are, however, of importance only with the third main vibration and generally are no more violent than the periodic forces of the next lower order acting in phase. If necessary, these exciting forces can be balanced within the engine. To get a clearer idea of these existing and natural frequencies they are shown on Figures 12 and 13, of which Figure 12 represents an installation of a ten cylinder oil engine with an elastic coupling between pinion and engine, whereas, Figure 13 is for a six cylinder oil engine and a hard connection between engine and pinion. In both cases only practically unimportant critical speeds remain in the operating range.

Conditions differ but very little when the number of pinions and oil engines working with one gear wheel is increased. The most important third main vibration remains as before because it is influenced almost entirely by flywheel and engine rotating masses of the individual oil engine.

The action of the exciting forces of two engines working with one gear drive, for instance, remains also practically the same, when the firing of both engines is synchronized, which can easily be done.

Experience with three motorships *Havelland*, *Munsterland* and *Vogtland* has completely proven the reliability of the type of installation used, an elastic connection between engine and pinion, and the hard connection of two engines to one gear drive as installed on *Monte Sarmiento* and *Monte Olivia* is equally justified by the service of the former to date.

It seems necessary at this time to refer to the originally projected elastic couplings of the engines of the *Monte Sarmiento* to their respective drive pinions by means of a quill shaft. Torsiograms taken on the primary shaft during dock trials registered unusually large pulsations of the turning moment at very low speeds while starting up with both engines working on

FIG. 12 AND 13.—EXCITING FREQUENCIES AND NATURAL FREQUENCIES OF OIL ENGINE GEAR DRIVE INSTALLATIONS.

Ten cylinder four cycle engine with elastic coupling between engine and pinion.

Six cylinder four cycle engine with hard connection between engine and pinion.

z = frequency exciting force.
• = critical speeds.
nI-nIII = natural frequency of mass elastic system.
a = No. of impulses per revolution of pinion.

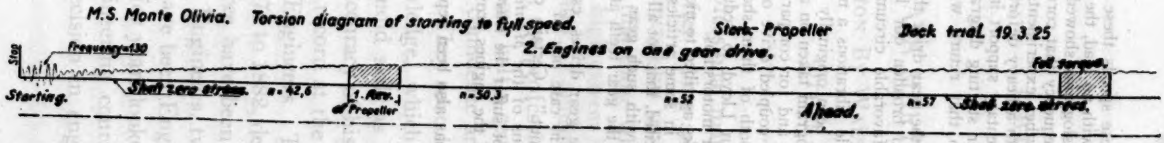

M.S. Monte Olivia. Torsion diagram of starting to full speed.
Stark-Propeller
Dock trial 19.3.25
2. Engines on one gear drive.
Stop
Frequency=130
Full torque.
Starting.
Shaft zero stress.
n=42,6
1. Rev. of Propeller
n=50.3
n=52
Ahead.
n=57
Shaft zero stress.

Fig. 14

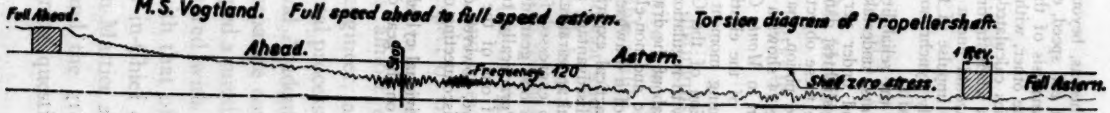

Full Ahead.
M.S. Vogtland. Full speed ahead to full speed astern.
Torsion diagram of Propellershaft.
Ahead.
Stop
Astern.
1 Rev.
Frequency 120
Shaft zero stress.
Full Astern.

Fig. 15

the same gear drive, whereas, beyond these speeds these torque variations became negligible and at full speed, even with no load, the installation justified all expectations. Analysis of the torsiograms showed that the engines oscillated, one against the other, with a frequency that corresponded to none of the natural frequencies calculated. Further experiments and calculations proved that both engines when starting with a heavy deformation (twist) of the long elastic shafts would find a momentary support in the mass of the gear drive and the first impulse of the air starting diagram would produce accelerations not exactly synchronous so that return waves could easily build up in the long shafts.

It is characteristic of the reliability of the gear drive that during various extraordinary experiments, undertaken to broaden our knowledge of the phenonema involved and under very unfavorable circumstances, the gear teeth did not suffer the slightest damage.

In order to overcome these objectionable vibrations a non-elastic connection between engine and pinion, that has been originally investigated, also, was resorted to. Figure 14 shows a torsiogram taken on a line shaft of the hard coupled installation in *Monte Olivia* and for comparison on Figure 15 is shown a torsiogram of the elastically coupled drive of *Vogtland*. The extraordinary even turning moment on both of them shows that both are entitled to the permission of the German Lloyd to reduce the line shaft diameter as compared with regulations for motorships.

Finally a few words about the drawbacks ascribed to mechanically coupled gear drives, both elastic and non-elastic, in some articles appearing in the literature. As a proof of a drawback is cited, that not all the critical speeds can be eliminated. Practical experience with such gear drives shows that they have no inherent disadvantage since the gear teeth at each examination show the same beautiful appearance.

Fears have been also expressed for the gear drives, especially when two engines are coupled to one gear drive, in case one of the pistons seizes. Even this has been "practically tried out." Certain circumstances contributed to a seizure of one of the pistons of the *Monte Sarmiento* on its first trip—the gear drive, however, did not suffer the least. In this case the possibility of quickly disconnecting one of the engines proved to be a decided advantage.

Disconnection of individual cylinders has also been repeatedly tried without the slightest effect on the gear drive.

OBITUARY.

ASA MARTINES MATTICE.

An Apreciation By Walter M. McFarland Assisted By Other Old Friends.

When a professional man of distinguished attainment has passed away, after a long and active life, it is natural that one who greatly admired his remarkable ability should endeavor to set it forth so that those who did not know him so well may understand the high esteem in which he was held by those who were his intimates. Counting from the beginning of his technical education, the professional life of Asa Martines Mattice covered just about half a century, and in every field of activity he enjoyed the confidence and respect of his colleagues in the highest degree. They soon learned to appreciate his great professional knowledge, which was almost encyclopedic in its scope, his splendid skill as an engineer, and the remarkable quickness and accuracy of his mental processes. His reputation began with his record at the Naval Academy where he led his class of Cadet Engineers. The separate course for Engineers lasting from 1871 to 1882, developed a number of very remarkable men. Three have been honored with the Presidency of the Mechanical Engineers, two with that of the Naval Architects and two have been Engineer-in-Chief of the Navy. All, however, have for years looked on Mattice as the ablest graduate of the engineering course, and the great majority have felt that his decision in engineering questions was the final word.

He entered the Naval Academy in the second class of Cadet Engineers, taking the two year course, and easily stood first during the whole time. His classmates tell of the remarkable ease with which he mastered his studies and there is no doubt that he was one of the few, like Admiral Taylor and the late Professor Spangler, who agreed with Professor Rankine in that celebrated little sentence of his "Applied Mechanics" reading: "Then by a simple and easy transformation we get the following equation." The experience of the vast majority of students of that monumental work, including the writer, has been complete disagreement with Rankine.

He graduated in 1874 and was at sea on the *Brooklyn*, *Vandalia* and *Trenton* until the summer of 1879. Admiral Willits, one of his classmates, tells a number of very interesting stories of his early days at sea of which the following is particularly interesting. While both of them were attached to the *Brooklyn* she went aground at Key West and there was natural anxiety as to the damage which might have been done to the propeller or the keel. There was a diving suit but no diver. The suit was very large, the waters full of sharks, and there were no volunteers among the enlisted personnel. Mattice, however, said that he would go down in the suit provided they would haul him up at once when he signalled one pull. So he went down, while the other officers stood around and wondered what would be the result. When Mattice had got to a depth of about ten feet there was a vigorous pull on the lanyard and the men handling the ropes pulled him up very quickly. The helmet was removed and Mattice said: "Nothing was the matter; I just wanted to see how quickly you could get me up." He then went down again and made an inspection which enabled him to make a very complete report of the conditions.

During the cruise on the *Vandalia* he had the pleasure of being a shipmate of General U. S. Grant, who was making a world tour after the conclusion of his term as President. Mattice has often told of the General's affability and simplicity

of character, and how he seemed specially to enjoy the company of the younger officers.

In September of 1879 he was ordered to duty at the Naval Academy where for the next three years he was an instructor in the Department of Steam Engineering. In association with the late Chief Engineer Kafer he developed a course in drafting and design which was the admiration of all who saw it, and the students who had the benefit of the course never ceased to speak of his high attainments as an instructor.

In 1882 he was ordered to sea on the *Miantonomoh* and the next year to the *Juniata*. In August 1883, the seismological instruments in Washington recorded a succession of terrific earthquake shocks which were estimated to center somewhere in the Dutch East Indies. As the *Juniata* was in Chinese waters and thus the nearest of our vessels to this locality, she was ordered south to ascertain the place and nature of the disaster and to render every assistance. It proved to be the great catastrophe of the eruption of Krakatoa, in the Sunda Straits, where about half of an island was blown to atoms, and the *Juniata* was the first foreign ship to bring succor. Besides the appalling loss of life and property, she found that all lighthouses, buoys and other aids to navigation had been destroyed. The *Juniata* remained in the vicinity for months, taking soundings, directing shipping, making new charts, etc. In this work Mattice had a leading part.

At the end of this cruise he was ordered to duty in the Bureau of Steam Engineering where he remained, except for a few months detached duty, during the rest of his active service in the Navy. This was the period of the beginning of what is generally called in the Service the "New Navy." Secretary Whitney continued the policy inaugurated by Secretary Chandler of building up the Navy with new and modern vessels. Chief Engineer, afterwards Admiral, Melville was appointed Senior Inspector at the Works of the Cramp Company who were building several vessels. The plans for some of these had been obtained abroad. Melville was satisfied that he could

produce designs equal to or better than these and made the proposition to Secretary Whitney. He was authorized to prepare the designs and Chief Engineer Kafer, Passed Assistant Engineer Mattice and Assistant Engineers Cathcart, Bull and Kaemmerling were ordered to report to him and assist him. Two of these officers have favored the writer with their recollections of the work done at that time. Professor Cathcart says: "Melville was an instinctive marine engineer of the first rank, Kafer was an accomplished designer with long years of experience, and Mattice was a living dynamo in his energy, speed and capacity for work, so the plans were completed in record time." It should be stated that there were no draftsmen to help these officers and all the work, both of calculation and drafting, was done by them. Cathcart adds: "In looking backward through the years to those torrid days of unremitting toil, the writer feels that Mattice,—with his tireless energy, his swift work and his vast fund of information—was a sort of *deus ex machina* in it all, supplementing fully Melville's genius and Kafer's surpassing skill." The design which was prepared at Philadelphia came into actuality as the machinery of the *San Francisco* and was the earliest of Melville's many successes in the design of machinery for our naval vessels.

In the summer of 1887 Melville was appointed Engineer-in-Chief of the Navy and Chief of the Bureau of Steam Engineering, and naturally took with him there Kafer and Mattice, who had been such splendid lieutenants in Philadelphia. Mattice was in charge of the drafting room and Chief Designer for the Bureau from August, 1887, to the end of December, 1888, when he went on leave for a year with the expectation of resigning, which he subsequently did in 1890.

While filling this position, Mattice produced the designs for the machinery of the original *Maine* (which was blown up in the harbor of Havana in the early part of 1898) and in connection therewith prepared the machinery specifications. These were the most elaborate and complete which had ever been drawn for any installation of marine machinery up to that time.

They served for many years with appropriate modifications as the model for all later specifications, as the writer knows from duty in the Bureau of Engineering after Mattice had left. Admiral Griffin, recently Engineer-in-Chief of the Navy for eight years, who was in the Bureau with Mattice, writes that "Mattice was responsible for the organization of the Bureau under Melville and this organization with slight changes has survived to the present time." He was a tireless worker and besides his regular stint at the office was busy with many other things. Those were the days, unfortunately, when there was not complete harmony between the different branches of the Service and an engineer of Mattice's ability was sure to take a prominent part in the controversies which took place. It would be quite interesting to tell about many of these because there were humorous as well as serious features. It was actually contemplated at one time to bring over a prominent English marine engineer to be the designer of our machinery. Mattice killed this by calling the attention of the newspaper correspondents to the fact that this would be against the "contract labor" law, and the matter was dropped. Mattice was absolutely fearless in upholding the dignity of his corps and profession and never allowed any thought of consequences to himself prevent his speaking absolutely plainly.

To those who are not fully cognizant of Naval history it may not seem that the work which has just been mentioned was remarkable because of the assumption that things in the Navy then were like they are now, where engineering is in the very forefront of progress. The facts are that, at the time Melville came into the Bureau, with Mattice as Chief Designer, engineering was at a rather low ebb. It had been so long since any new vessels were authorized, that Naval officers had had no intimate experience with the latest designs. That Mattice was able to do such splendid work is explained in a note of Professor Cathcart;—"Although his career was largely one of action, he was fundamentally a student, and, from boyhood till death came, his search for knowledge was unceasing. With this

desire, was joined a memory that very seldom failed, so that, as the years passed, his information on engineering subjects became encyclopedic." In other words, although the machinery in service was not of the latest type, Mattice had been preparing himself and, when the occasion offered, was ready to step in and do work of the highest class.

Before ending the reference to his Naval career, it is pleasant to quote from one of the young officers who was with him at Philadelphia, Assistant Engineer (now Admiral) Kaemmerling;—"One thing that impressed me with Mattice more than any other man of marked ability with whom I was ever associated is worthy of notice. In spite of preeminence, he was quick to accept anything a subordinate might suggest, if it had any value, and thus gave the greatest incentive to work with him." This readiness to listen to others, especially where their advice might be of value, was eminently characteristic and lasted throughout life.

When he left the Bureau of Steam Engineering, he joined the staff of E. D. Leavitt of Boston as his principal assistant. Mr. Leavitt enjoyed a high reputation as a consulting engineer and his work was of the highest class. A great deal of it was for the Calumet & Hecla Mine. Mr. George H. Cox, who was in charge of the financial end of the office, has recently written that, during the time he was with Mr. Leavitt, Mattice really was responsible for everything that went on, as Mr. Leavitt was getting along in years, although, of course, he exercised supervision. The writer happens to know that Mattice enjoyed the fullest confidence of President Agassiz and Vice-President Livermore of the Calumet & Hecla Mine.

Mr. W. H. Glocker, for many years Superintendent of Shops at Cramp's Shipbuilding Company, was Chief Inspector for Leavitt during Mattice's term there, and has kindly supplied the following information:—

"It was with deep sorrow and regret that I learned of the death of our old friend, Asa M. Mattice. My association with him was while he was with E. D. Leavitt for about ten years

from 1889. I considered him without exception the best all-around engineer of my acquaintance. He was a hard and untiring worker with a capacity for work that was simply marvelous. During his stay in the Cambridge office he had full charge of all designing, and he turned out plans for hoisting and pumping engines and mining machinery that remain unsurpassed for both quality and size.

"Among the machinery built from these designs are the following:—five of the largest triple-expansion hoisting engines and their hoisting gears; the four-cylinder, triple-expansion, air compressor engine *Mackinac* and the engine *Wabeek* for the Calumet & Hecla Mining Company; the 72,000,000 gallon sewage pumping engine at the Calf Pasture Station and a large water-supply pumping engine at Chestnut Hill Pumping Station for the City of Boston; the large pumping engine at Fresh Pond Pumping Station for the City of Cambridge; the 15,000 H.P. pumping engine for the hydraulic, armor-forging press for the Bethlehem Steel Company. Besides these there were other important machinery installations for Boston and New Bedford, for the American Tube Works, the Pope Tube Company and others while Mattice had charge of the Leavitt office. This machinery was all of the finest type and the greater part, the largest of its kind."

His close friend, Mr. Charles C. Tyler writes, with reference to this period of his life, as follows:—

"It is my impression that Mattice's work, while with Leavitt, represented his highest achievements in engineering, and was the part in which he took the greatest pride. I visited the Calumet & Hecla mining properties in 1904 or 1905 with Mattice, and saw the most remarkable hoisting, pumping and other mining machinery I have ever seen anywhere. It was not designed and built as an ordinary commercial product on a competitive price basis. The governing factors were absolute reliability and continuous service. These results were secured, and the machinery gave complete satisfaction for years in a country far from sources of supply and where quick repair was

impossible. Few concerns could afford such machinery as the initial cost was high, but for such an enterprise as the Calumet & Hecla, who could figure for a long term of years, it was true economy. This machinery was installed before the introduction of electrical apparatus on any large scale, and the economy of its maintenance and operation was remarkable."

After about ten years of splendid work with Leavitt, Mattice left him and was occupied in consulting work alone. He once told the writer that his reason for leaving was that Mr. Leavitt was growing old and was somewhat difficult to get on with. They had always been the best of friends and still were and he wanted to leave before anything occurred that might cause a change.

A pleasant feature of his life in Boston, and one which illustrates a side of his character which was not appreciated by those who only knew him slightly, was the organization of the New England Society of Naval Engineers. This was entirely Mattice's work. He found that there were a good many former officers of the Engineer Corps, particularly those who had served during the Civil War, resident in New England. With a great deal of trouble and painstaking care he tried to find every man who was eligible to such an organization and finally brought together about one hundred and twenty-five members. The Society was a social affair and only met once a year at an annual dinner about Thanksgiving Day.

Dr. Hollis, President of Worcester Polytechnic Institute says:—"It had been going two years when I went to Harvard and when I joined one hundred and twenty-five of us sat down together in Young's Hotel under Mattice's amusing contributions. There is a feeling of sadness when I think of these dinners, because I went to them even after I came here to Worcester and every one has dwindled in size. If I am not wrong, Brooks and I are now the only survivors of the long past. Mattice was one of them and I saw him at the last dinner which was held in Boston a year or two ago."

Dr. Hollis in writing about Mattice adds;—"I entirely agree with you about Mattice. I think he was a man of very remarkable intellectual qualities and really in many ways a great engineer. As a master of the technicalities of engineering I doubt if he could have been surpassed in the United States. I saw him frequently while he was with Leavitt and we often dined together."

The characteristic which is brought out by the dinners of this Society is the fact that he was a very affectionate, although not a demonstrative man. Some years after he left Boston, while he was in Pittsburgh, he went to these dinners, traveling about 750 miles each way for the pleasure of being with these old friends. It is sad to relate that, due to his train being late, he entirely missed one of the dinners, and for the same reason he arrived at a second one just as the diners were about to disperse. I knew also of another occasion, when there was a reunion of some members of his class in Washington, and he came all the way from Boston to attend the dinner, arriving only an hour or so before and taking the midnight train back after it was over. Those of us who had the privilege of intimate association with him realized how strong was this affection for old friends.

After severing his connection with Leavitt's office, he spent something over a year in consulting work and then joined the Westinghouse Electric & Manufacturing Company as Chief Engineer. During his incumbency many new lines of product were developed, including the great generators for the elevated and subway systems in New York; the first very large turbo-driven generators (5000 K.W.); street railway motors; new lines of standard D. C. and A. C. motors; electrical instruments, switchboards, etc.

After about three years with that Company Mr. Westinghouse transferred him to the Westinghouse Machine Company as Chief Engineer, believing that his services were more needed and could be better utilized in that Company; he also retained a connection with the Electric Company as Consulting Engi-

neer. It was during this period of nearly four years that the writer was closely associated with him while himself connected with the Westinghouse interests. The story here is the same as everywhere else in his career, of unremitting devotion to work and the maintenance of the highest standards in all designs. While an absolute master of the theory of engineering he was strongly practical and one feature of his methods has always appealed to me very strongly. When a new design for a line of apparatus had been prepared in the Drafting Room, before putting it in the shop, he always called in the foremen of the different Departments who would have to handle it, for inspection and criticism of the design, with a view to arranging as far as practicable that the work in each Department should be free from unnecessary difficulties, with the consequent expense. He even carried this so far that in some cases he had skeleton models made to be sure that no parts conflicted. He could read a drawing as well as anybody but experience had taught him that the time to avoid difficulties is in the preliminary stages and not after castings are made and work is begun. It is pleasant to relate that this action met with very hearty co-operation from the foremen who appreciated the compliment to their judgment in being called upon to pass on the designs.

Mattice's relations with the Westinghouse Company were entirely pleasant, but early in 1904 his old friend and classmate Mr. B. H. Warren became President of the Allis-Chalmers Company and urged Mattice to come with him as Chief Engineer and in charge of manufacturing. The call of friendship prevailed and he became Chief Engineer of the Allis-Chalmers Company in April of 1904. That Company had determined upon important additions to its diversified products; notably the Fullagar-Parsons steam turbine, turbo-generators, hydraulic pumps, etc., and also to re-design and modernize many of its old lines of product and those of the Bullock Electric Company, which it had recently acquired. He had been

associated while with the Westinghouse interests with Mr. Charles C. Tyler with whom he was again associated in the Allis-Chalmers Company where Mr. Tyler was General Superintendent. Together they planned the large extension of the Allis-Chalmers works at Milwaukee which were built at this time and were thoroughly modern and up-to-date in every way.

Some of these additional products had been developed in England and on the Continent. Mattice made several trips to familiarize himself with the state of the art there, and thus was able to expedite the development of apparatus, adapted to American practice, without a long period of experimentation. The great diversity of apparatus produced by the Allis-Chalmers Company and the engineering problems involved, required the expenditure of such intensive effort that after two years of successful service with that Company, Mattice resigned to become a partner in the newly organized firm of consulting engineers, Kafer, Warren and Mattice, with offices at 60 Wall Street, New York.

The professional experience of the three members of this firm had covered a wide range and they had such an extensive acquaintance with engineering enterprises that the work entrusted to them was important and growing in volume, but Mr. John C. Kafer died in March, 1906, and Mr. B. H. Warren in October, 1906, both within a year after the organization of the firm. They had been intimate friends of Mattice from early manhood, and their death so affected him that he felt unable to continue the business alone. In 1907 he became Works Manager of the Walworth Manufacturing Company of Boston, Mass. Here he had full charge of the design and manufacture of its many products, and he continued in this position until the close of 1911.

In 1911 he retired from active business, but a man of his energy could not be idle. To keep himself occupied and to lead an open air life, he purchased a farm near Lockport, N. Y., where he raised fruit and chickens. This was obviously a great change from engineering, but the careful planning, which

had been his work for years, came in here as well, and some very interesting stories could be told of things that he did on the farm which might be called the application of engineering ideas to farming.

Meanwhile his old friend Mr. Tyler had become connected with the Remington Arms Company and had reached the position of Vice-President. When the Great War began and the Allies applied to our establishments for munitions, the demand upon the Remington Company was of course enormously increased. Mr. Tyler was in charge of the shops and their operation and found that the extensions which were necessary, and the supervision required, were more than he could personally manage. He besought Mattice to give up farming and come back into engineering to help him, because he knew that any work which was turned over to Mattice required no further supervision. Mattice consented and joined the Remington Company in 1915, with which he continued until his death, his position being that of Advisory Engineer.

The Company had contracts with the several Allied Governments for small arms, and ammunition therefor, in huge quantities, and later similar contracts for still greater varieties and greater quantities of similar munitions with the U. S. Government. Mattice's work in connection with these contracts was invaluable to both parties as it simplified and made more understandable the intricate and technical contract specifications for the manufacture, inspection and acceptance of the products.

After the Armistice he prepared most of the data required by the Company for the settlement of its war contracts with the United States and other Governments, and also in connection with its several Government suits. His work was so carefully and thoroughly performed as to win the admiration, as well as the approbation, of the legal representatives of both parties to the suits.

During 1917 and 1918 Mattice was Secretary of the Small Arms and Ammunition Manufacturers Production Committee, whose members comprised representatives from each of the

manufacturers of these products, who had contracts with the U. S. Government. Its sessions were held monthly and its work was most important to the Government, who sent many officers from the Procurement, Production, Inspection and Engineering Divisions to attend the meetings of the committee, and advise as to the existing needs and probable future requirements of the Ordnance Department. The records kept by Mattice were most comprehensive and proved very helpful to the committee members and Government officers.

While he was associated with Leavitt he made several trips to Europe to inspect the latest practice and improvements. He was also sent abroad by the Westinghouse Company on important work for them. His sense of humor was very keen and many clever stories resulted from these trips, especially those to the Continent of Europe where customs are so different from ours.

His mind worked so quickly that he could detect instantly errors and absurdities in a newspaper story that the average reader, even if a good engineer, would pass over from less acuteness of mind. He used to refer to such cases as "newspaper arithmetic" and he had much amusement in developing the data given so as to show how absurd they were.

He had been a member of the American Society of Mechanical Engineers since 1889 and was a Manager 1903-1906. He was also an original member of the Society of Naval Engineers and was a member of The Engineers' Club of New York.

He was born in Buffalo, N. Y., August 1, 1853, and was the youngest of a large family of children, of whom only one sister, Miss Cora Mattice, remains at the advanced age of 85. He had never married. He died at The Engineers' Club, New York City, April 19, 1923, in his seventy-second year.

ASSOCIATION NOTES.

CHANGE IN COUNCIL.

Commander H. S. Howard (CC), U. S. Navy, having resigned as a member of the Council because of detachment from duty in Washington, the Council has appointed Lieutenant Commander E. L. Cochrane (CC), U. S. Navy to fill the unexpired term of Commander Howard.

MEMBERSHIP.

The following have joined the Society since the publication of the last JOURNAL:

NAVAL MEMBERS.

Adams, R. M. B., Ensign, U. S. Navy.
Beehler, W. P., Commander, U. S. Navy (Retired).
Bowers, R. W., Lieutenant, U. S. Navy.
Christie, Harlie H., Rear Admiral, U. S. Navy.
Day, Geo. C., Rear Admiral, U. S. Navy.
de Steiguer, Louis R., Rear Admiral, U. S. Navy.
Dayton, John H., Rear Admiral, U. S. Navy.
Hough, H. H., Rear Admiral, U. S. Navy.
Jackson, R. H., Rear Admiral, U. S. Navy.
Scales, A. H., Rear Admiral, U. S. Navy.
Senn, T. J., Rear Admiral, U. S. Navy.
Vogelgesang, C. T., Rear Admiral, U. S. Navy.

CIVIL MEMBERS.

Haas, Albert F., care New England Steamship Co., Newport, R. I.

ASSOCIATE MEMBERS.

Gustafson, Walter A., 124 Southern Avenue, Dorchester, Mass.

Knothe, A., Engineer Commander, R. British Navy.

le Page, G. W., Engineer Commander, R. British Navy.

Lobera, Miguel U., Lieutenant Engineer, Argentine Navy.

Nash, Samuel A., 116 Westford Avenue, Springfield, Mass.

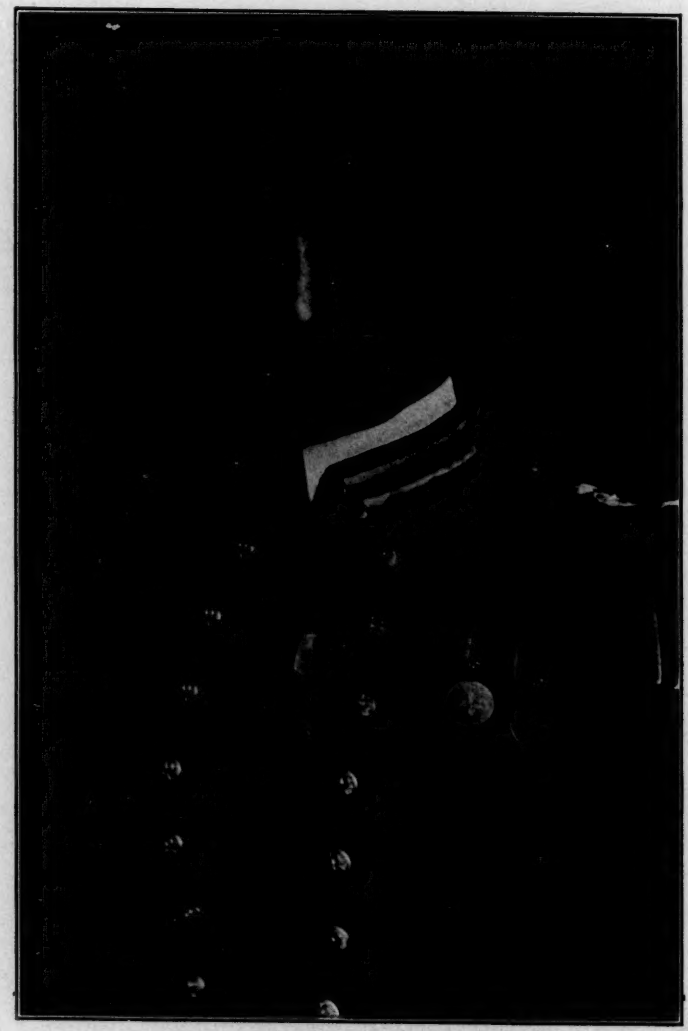
Patterson, Oliver W., 429 Wrightwood Ave., Chicago, Ill.

Porter, Frederick P., 814 E. 228th St., New York, N. Y.

Ray, W. R., 105 Hayward Avenue, San Mateo, Calif.

Rickard, S. D., Preferred Utilities Co., 33 W. 60th St., New York, N. Y.

Stine, Wilmer E., 611 Fairview Avenue, Brooklyn, N. Y.



CHARLES WILSON DYSON
REAR ADMIRAL, UNITED STATES NAVY.

REAR ADMIRAL DYSON.

On December 2, Rear Admiral C. W. Dyson retires for age and with over 46 years active service. He graduated from the Naval Academy in 1883. He served at sea on the U. S. S. *Adams,* U. S. S. *Fish Hawk,* U. S. S. *San Francisco,* U. S. S. *Machias,* U. S. S. *Massachusetts,* U. S. S. *Monterey,* U. S. S. *Raleigh* and the U. S. S. *Oregon,* and also as Fleet Engineer of the Asiatic Fleet. His shore duty was at the Mare Island Navy Yard, Boston Navy Yard, the U. S. Fish Commission at Washington, the New York Shipbuilding Company and the Bureau of Engineering, Navy Department. He served on the U. S. S. *San Francisco* during the Spanish War and was in the Bureau of Engineering during the World War.

He was awarded the Distinguished Service Cross for his work in the Bureau during the World War. This award was unusually appropriate, for it defines very accurately the type of service—distinguished—that he has always rendered, not only during the World War but at all other times during his Naval career. His work has been recognized, not only by the American Government, but also by foreign countries as well. He is one of the few American naval officers who has an international reputation in engineering. His work on "Screw Propellers" is recognized as a classic all over the civilized world.

In addition to his regular naval work he has served at various times as President, as Secretary-Treasurer, and as a Member of the Council of the American Society of Naval Engineers. He also has made numerous contributions to the JOURNAL of this Society. His name, as inventor, is connected with almost every type of apparatus used in marine engineering, such as boilers, evaporators, feed heaters, combined oil and powdered fuel burners, etc.

Fortunately he expects to continue active engineering work after retirement and the marine engineering world will continue to profit by his services.

ASSOCIATION NOTES.

MEMBERSHIP.

The following have joined the Society since the publication of the last JOURNAL:

NAVAL MEMBERS.

Adams, R. M. H., Ensign, U. S. Navy.
Beelder, W. P., Commander, U. S. Navy (Retired).
Bowers, R. W., Lieutenant, U. S. Navy.
Christie, Harlie H., Rear Admiral, U. S. Navy.
Day, Geo. C., Rear Admiral, U. S. Navy.
de Steiguer, Louis R., Rear Admiral, U. S. Navy.
Dayton, John H., Rear Admiral, U. S. Navy.
Hough, H. H., Rear Admiral, U. S. Navy.
Jackson, R. H., Rear Admiral, U. S. Navy.
Scales, A. H., Rear Admiral, U. S. Navy.
Senn, T. J., Rear Admiral, U. S. Navy.
Vogelgesang, C. T., Rear Admiral, U. S. Navy.

CIVIL MEMBERS.

Haas, Albert F., care New England Steamship Co., Newport, R. I.